POWER

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Strikes

THE land is full of strikes and rumors of strikes.

Fuel and transportation are two of the main essentials of human existence upon the present scale.

A sustained cessation of the production of coal, a serious interruption of the transportation system means more to this country than a foreign invasion.

A Government that can seize property and draft the flower of the youth of the country to stand up and be shot for the common good ought to be able to find out what this internecine row is about and settle it or compel its settlement in some way that will not involve the long-suffering non-belligerent public.

Labor ought to have a wage upon which it can live, not only from hand to mouth but which, with thrift and industry, will permit it to tide over periods of misfortune and take care of itself when its productive period is over. It should have continuity of opportunity and conditions as sanitary and agreeable as the nature of the employment permits.

The employer of labor should have a fair return upon capital actually invested, a fair salary for his own labor, if he performs any, and a fair profit upon the turnover of the undertaking in which he has risked his money and upon which he has expended his brains, initiative and enterprise.

What do either of them want more than this?

The employer says that tne laborer wants an exorbitant wage for a slim day's work, that he insists upon the unnecessary division of simple jobs among a multiplicity of trades, requires an unnecessary number of men to do a job just for the purpose of creating work for others and decreasing the available labor supply: all of which makes the products of labor more expensive to everybody, including the laborer himself. That labor, when organized, autocratically interferes with the discipline and management of the works through walking delegates and other intruders.

The laborer says that no matter how much he may, by application and efficiency, reduce the cost of production of the necessities of life, the producer will sell them, not at cost plus a fair profit but for what he can get for them, and that through trusts and holding companies and comity of interest big business is organized to make this as much as the traffic will bear. He says that executive salaries are not proportioned to the importance of the contribution of their recipients to the service rendered, but are fixed by the holders of the control, and that such salaries and profits are as sacred from public inspection as the intimacies of their ladies' toilet.

If the employers and the employees of the great essential industries of the country cannot settle these things between them without upsetting the whole social and industrial system, the big Public can and will; and the first thing that it will go after in its determination of the right of the matter will be the Facts involved.

How much of the price paid for a ton of delivered coal goes to labor and how much to somebody else. When the public knows this it will know where justice lies and what to do about it.

Queenston-Chippawa 55,000-Horsepower Hydraulic Turbines*

Five Units Being Installed To Operate Under 305-Ft. Effective Head—Each Weighs 2,000,000 Lb., 620,000 Lb. of Which is in the Turbine—New Type of Turbine Casing and Operating Gear Developed — Special Control Stand for Operation of Machines.

BY FRANK H. ROGERS
Hydraulic Engineer, William Cramp & Sons Ship and Engine Building Company

WHEN in 1919 the Niagara Falls Power Co. installed in its new station No. 3 extension, hydraulic turbines each of 37,500 hp., described in *Power*, Sept. 14, 1920, these machines were the most powerful units ever built, and they maintained this record until Dec. 28, 1921, on which date the Queenston-Chippawa plant of the Hydro-Electric Power Commission of Ontario was officially opened and the first 55,000-hp. turbine started in operation. These turbines are now the most powerful units in the world by a margin of nearly 50 per cent.

As explained in the article in the June 27 issue, the power house is near Queenston on the Canadian side of the Niagara River, about four miles below the Horseshoe Falls. The intake is located near Chippawa, about two miles above the Falls. From the intake the water flows through a portion of the Welland River and through a canal 8½ miles long to the forebay, where the elevation of the water is about 308 ft. above the water level in the river below. The present installation consists of five 55,000-hp. turbines, two of which are furnished by the Wellman-Seaver-Morgan Co., three of the turbines and the five governing mechanisms by the I. P. Morris Department of the William Cramp & Sons Ship and Engine Building Company.

*Abstract of paper presented at the Hydro-Electric Conference held under the auspices of the Philadelphia Engineers Club, March, 1922.

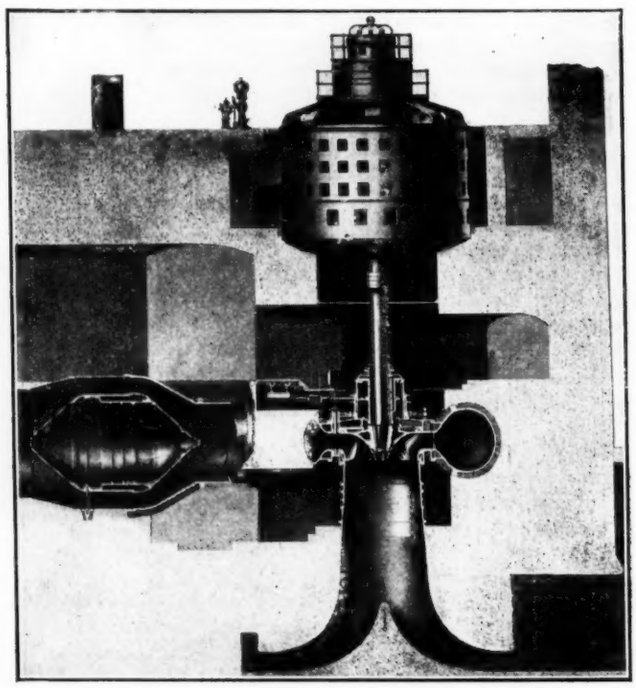


FIG. 1. SECTIONAL VIEW OF 55,000-HP. TURBINE AND OUTSIDE VIEW OF GENERATOR

FIG. 2. TURBINE ASSEMBLED IN THE SHOP

Fig. 1 shows a sectional elevation through one of the I. P. Morris turbines and Johnson valve and an outside view of the generator. Each turbine is designed to develop 55,000 hp. under an effective head of 305 ft. at a speed of 187½ r.p.m. The quantity of water required at full load will be about 1,750 cu.ft. per sec. These units are of the vertical shaft, single-runner type, provided with a cast-steel runner, cast-steel volute casing and Moody spreading draft tube formed in the concrete substructure of the station. The great size of the unit will be appreciated from the following weights:

The total weight of the generator is 1,380,000 lb. and of the turbine 620,000 lb., giving a total weight for the combined unit of 2,000,000 lb. The thrust bearing supported above the generator is designed to support a total revolving weight plus allowance for hydraulic thrust of nearly 1,000,000 lb. For the first three units Kingsbury bearings were furnished and for the two remaining units, General Electric spring bearings.

Owing to the great size of these units and the high head under which they operate, special construction was required in order to avoid excessive stresses and prevent leakage. Therefore the usual design of volute casing with separate speed ring was not considered the best, and a new type was developed especially for this installation, known as the Taylor sectional spiral casing.

Fig. 2 shows one form of this new type of casing, and Figs. 3 and 4 indicate how the radial joints are bolted in the new and the usual types of casings. In the new design, Fig. 2, the casing is divided into a number of radial sections. The larger radial sections are subdivided into two parts, the inner section being cast with the stay vane and connected by longitudinal flanges to the outer volute section. In previous designs a separate speed ring was provided, containing the stay vanes, and was bolted to the casing sections through horizontal flanges, as indicated in Fig. 4. This results in a long unsupported section across the throat at the

radial flanges, causing excessive loads on the bolts and consequent leakage. In the new design the radial flanges are extended to the inner bore of the casing, as indicated in Fig. 3, thus increasing the number of bolts and at the same time decreasing the unsupported

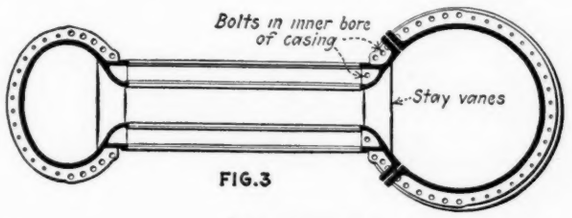


FIG.3

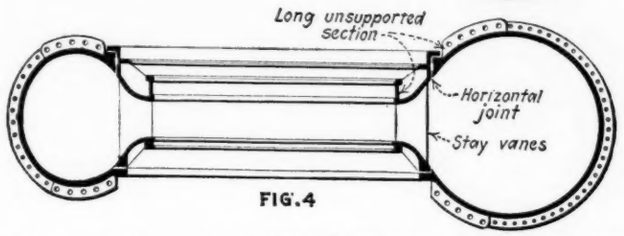


FIG.4

FIGS. 3 AND 4. INDICATE, RESPECTIVELY, HOW THE RADIAL JOINTS ARE BOLTED IN THE NEW AND THE USUAL TYPES OF CASING

section, resulting in low stresses and a casing free from leakage. The casing is made in 12 sections, the total weight being about 290,000 pounds.

The casing was tested in the shop to a hydrostatic pressure of 260 lb. per sq.in., this being double the normal operating pressure. When the pressure was

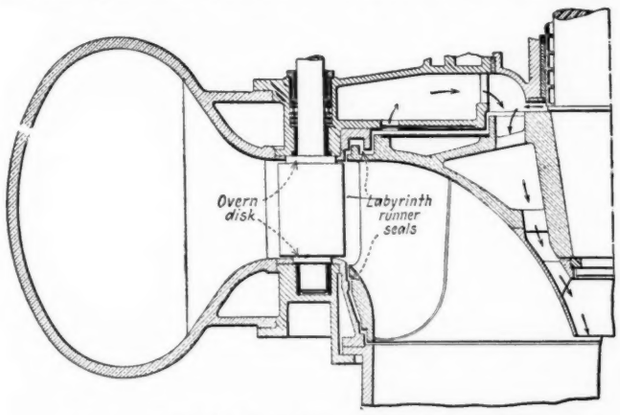


FIG. 5. SECTION THROUGH TURBINE SHOWING LABYRINTH RUNNER SEALS

applied for the first time in the shop, the casing was found to be absolutely tight against leakage. The total load on the head bolted to the casing intake flange during the test was 3,000,000 pounds.

In order to develop high efficiency and to maintain this efficiency for a long period, a number of special features were adopted for the various parts. The design of runner adopted is practically the same as that used in the 37,500-hp. turbine built for station No. 3 extension of the Niagara Falls Power Co., as this runner not only developed a high maximum efficiency in actual operation, but also showed an efficiency of over 90 per cent from about half to full load. This performance was in part due to the labyrinth seals adopted for the runner, as shown in Fig. 5. The leakage water in its path from the discharge of the guide vanes to the draft tube must pass through three separate seals, in each of which the velocity head is destroyed. As a result the head producing leakage at each seal is only one-third of the head existing at the discharge of the guide vanes. It is estimated that at full load the leakage at the seals of this runner amounts to 14 cu.ft. per sec., which is only 0.8 of 1 per cent of the full-load quantity. If this runner had been provided with the usual design of single seal, the leakage would have amounted to about 26 cu.ft. per sec., or 1.5 per cent of full-load quantity.

In Fig. 5 is also shown the passage of the leakage water from the upper runner seals through the head cover and vent holes in the runner to the top of the draft tube, which results in minimum hydraulic thrust on the runner. This method of reducing the leakage results not only in high initial efficiency, but also in the maintenance of this efficiency for a long period of time, for, owing to the much lower velocity of the leakage water the wear at the seals will be materially reduced.

The runner is of cast steel, the vanes being cast integrally with the crown and band, Fig. 6. The outside diameter is 10 ft. 4⅞ in., and its weight is 42,000 pounds.

The guide vanes are of the Overn disk type, as indicated in Fig. 5. Heavy disks are cast between the vane proper and the upper and lower shanks, as indicated in the figure. These disks, when in place, are flush with the upper and lower distributor plates, so do not interfere with the flow of water through the vanes. The disks reduce the amount of leakage water passing above and below the vanes and guide the water at the proper angle to the runner. This not only results in higher efficiency, but also materially reduces the leakage when the guide vanes are closed. In addition to these points this type of guide vane results in a stronger mechanical connection between the vane and the shanks.

For the three I. P. Morris turbines the draft tubes are of the Moody spreading type and not only regain the axial components of the discharge water from the runner, but also regain the whirling components, result-

FIG. 6. THE RUNNERS ARE OF CAST STEEL, THE VANES BEING CAST INTEGRAL WITH CROWN AND BAND

ing in high part gate efficiencies and operation of the unit free from the usual vibrations.

In the case of the two turbines furnished by the Wellman-Seaver-Morgan Co., No. 1 unit is provided with a curved draft tube and No. 2 is a Moody spreading draft tube, as in Fig. 1. As these two units are of exactly the same design throughout with the exception of the draft tubes, the efficiency tests, which will

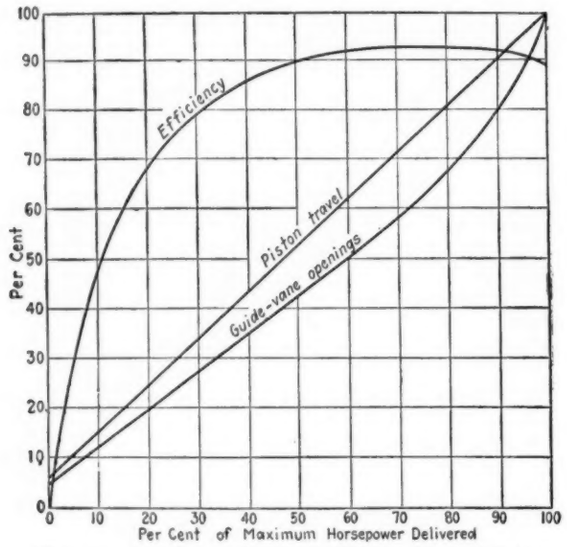


FIG. 7. GUIDE-VANE OPENING AND PISTON TRAVEL REQUIRED FOR UNIFORM SPEED REGULATION

probably be made at an early date, will be of special interest in showing the difference in efficiencies of the two turbines resulting from draft-tube design.

The speed regulation of units of so great a power is of vital importance, and for this reason a special operating gear was adopted. In order to obtain a uniform degree of speed regulation at all points of operation, a given load change on the generator should result in the same speed change irrespective of the initial load on the unit. In turbine governors the design is usually such that a given speed change always results in the same amount of travel of the piston of the operating cylinder for all positions. If, therefore, the operating gear of the turbine is designed so that a given piston travel produces the same change in power on the turbine, the desired result is accomplished; namely, the same speed change for a given load change at all points of operation.

To study the requirements for the 55,000-hp. turbines, consider the curves shown in Figs. 7 and 8. In Fig. 7 the horsepower delivered is plotted against the efficiency of the turbine and the guide-vane opening. The desired piston travel is also plotted as a straight line in order to meet the requirement that a given piston travel produce the same increment of power. It will be noted that, owing to the shape of the efficiency curve, the guide-vane-opening curve is concave from above, the increments of guide-vane-opening increasing as the horsepower increases, particularly beyond the point of maximum. At 90 per cent horsepower the guide-vane opening is only about 80 per cent, where 100 per cent guide-vane opening is required for 100 per cent horsepower. The curves, Fig. 8, show the piston travel, plotted directly against the per cent guide-vane opening to obtain the desired straight-line horsepower curve. To obtain a uniform degree of speed regulation at all points of operation therefore, the operating gear should be designed to produce a guide-vane-opening curve with respect to piston travel as shown. For mechanical reasons it is usually impossible to produce this curve exactly, but the special type of operating gear developed by the writer approaches quite closely to the desired results. This gearing is known as the "offset-lever gate-operating mechanism."

In Fig. 9 is a plan view of this new operating gear, and it will be noticed that the levers attached to the stems of the guide vanes are offset from the usual radial position. To close the turbine gates, the operating ring turns in a counter-clockwise direction when viewed from above, so that as the gates close, the levers are moved farther away from the radial position. In this gear, therefore, the links connecting the levers to the operating ring are in compression when the gates are closing. For a given movement of the operating ring, the amount of guide-vane movement will decrease as the turbine gates close and this relation may be expressed by the formula,

$$\frac{\textit{Angular velocity of vanes}}{\textit{Angular velocity of operating ring}} = \frac{\textit{Sin } R}{\textit{Sin } V}$$

In which R is the angle between the link and the radius of the operating ring and V the angle between the link and the radius of the vane lever.

As the guide vanes close this ratio decreases, thus resulting in a decreasing angular velocity of the guide vanes; that is, a decrease in the amount of guide-vane movement.

In Fig. 10 the curves are plotted to show the comparison between the usual radial-lever gate mechanism and the offset-lever gate mechanism as applied to the 55,000-hp. turbines. The piston travel is plotted

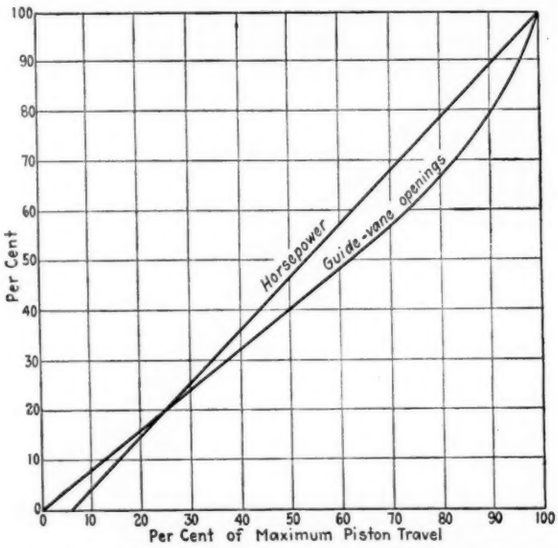


FIG. 8. PISTON TRAVEL AND GUIDE-VANE OPENING TO OBTAIN THE DESIRED STRAIGHT-LINE HORSEPOWER CURVE

against guide-vane opening, and it will be noted that the offset-lever type results in a curve having a slight concavity from above, whereas the usual radial-lever type results in a curve having a slight convexity from above. This shows that for the offset-lever type the increment of guide-vane opening increases as the piston opens the gates, whereas in the case of the radial-

lever type the increment of guide-vane opening decreases.

Another point to be considered in the design of the operating gear is the fact that at small gate openings on a turbine the hydraulic conditions are poor, resulting in eddies and whirls which cause internal load changes, which vary rapidly. These internal load changes have no connection whatsoever with the load changes on the generator. Such internal load changes, however, result in rapidly fluctuating speed changes and hence movements of the governor, which in turn produce movements of the operating piston. It is therefore desirable to reduce, as far as possible, the effect of these speed changes on guide-vane movements. With the offset-lever type, the guide-vane movements are smaller at low-gate openings than in the case of the usual radial-lever type. This new gear should therefore considerably reduce hunting of the governor, which is so frequent a trouble at small-load conditions on a unit, and it should be of particular value at no-load speed when synchronizing the generator.

Another advantage of the new type of gear is the possible reduction in the governor capacity. This value is fixed by the maximum hydraulic load on the guide vanes, which always occurs at the closed position. With the new type of gear the torque on the vane levers increases as the gates close, becoming a maximum at closed position, where required. In the case of the 55,000-hp. turbines, the governor capacity is 230,000 ft.-lb., whereas if the usual radial-lever type gear were used, the governor capacity would have to be increased to 260,000 ft.-lb. This lower governor capacity not only reduces the size of the operating cylinders, but also the size of the governor pumps, accumulator tanks, sump tanks and piping.

As the links connecting the operating ring to the guide-vane levers are in compression when closing, it is necessary to design these links so that they will break in compression in case an obstruction lodges between two of the guide vanes, and thus protect the

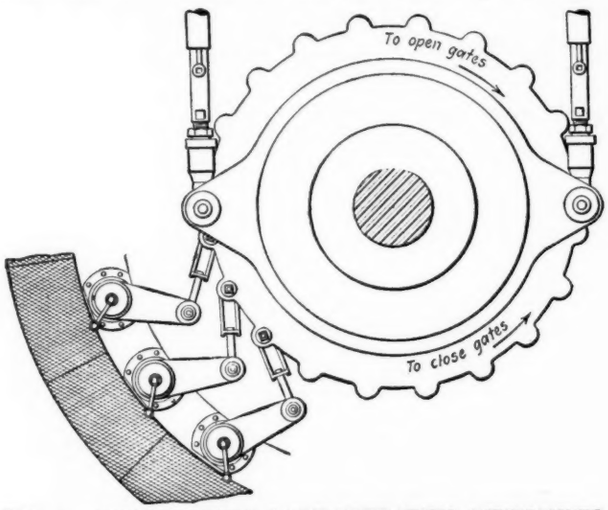


FIG. 9. OFFSET-LEVER GATE-OPERATING MECHANISM

remainder of the operating gear. The design of one of these links is shown in Fig. 11, from which it will be seen that one end of the link consists of a single bar and the other end of a double bar, the two ends being cast together through a transverse section, which will break in shearing. These links will also protect the operating gear for opening movements by breaking in shearing if an obstruction should become lodged between the back of the guide vane and the casing stay vane.

Special precaution was taken in the design of the lignum-vitæ guide bearing in order to insure an ample supply of water to this bearing under all conditions. Cored passages are provided on the sides of the bearing so that the lubricating water is admited not only at the top of the bearing, but also at two intermediate points between the top and bottom.

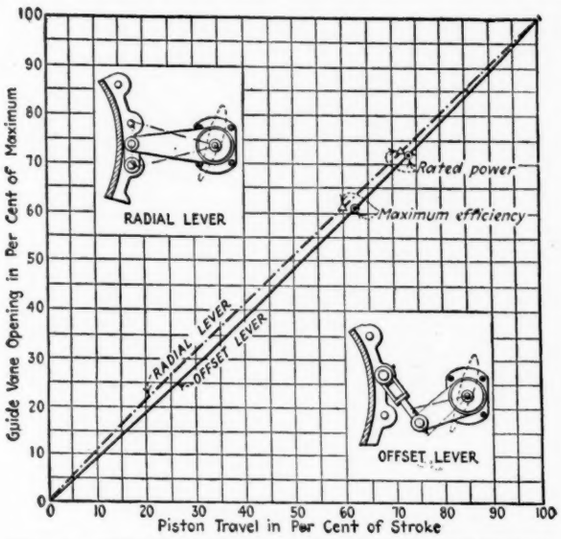


FIG. 10. CURVES SHOWING COMPARISON BETWEEN RADIAL-LEVER AND OFFSET-LEVER GATE MECHANISM

The upper sections of the draft tube are made of cast iron, the center section being cast in four pieces, so that it can be removed when it is desired to dismantle the runner from below without interfering with the generator.

The five governors furnished for this plant are of the I. P. Morris double-floating lever type. The governor actuator and hand control are located on the main power-house operating deck. These governors are furnished with the Taylor automatic control for changing over from governor to hand control or vice versa. To accomplish this result, pressure-operated plunger

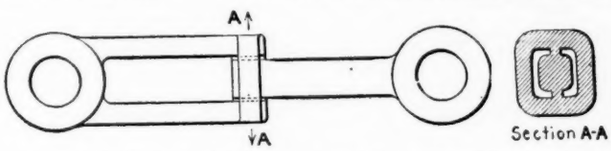


FIG. 11. SHEARING LINK FOR OFFSET-LEVER MECHANISM

valves are placed in the pipes from the main-governor valve and similar smaller plunger valves are placed in the hand-control stand. A single movement of a lever will close simultaneously all the plunger valves on the hand control and open those on the governor. Therefore the change over can be made in a fraction of a second and insures proper sequence of valve operation without danger of losing control of the unit. The main governor valves are on the turbine operating deck.

A central pumping system is installed, consisting of two motor-driven pumps, each having a capacity of 700 gal. of water per minute against a pressure of 200 lb. per sq.in. Each of these pumps is designed to

take care of five main units, the second pump being held as a spare. The governor fluid is supplied from a central concrete sump tank and delivered by the pumps into a common pressure main, which is connected to the accumulator tank at each unit. The return water from each governor is delivered into a common return main which empties into the concrete sump tanks. The

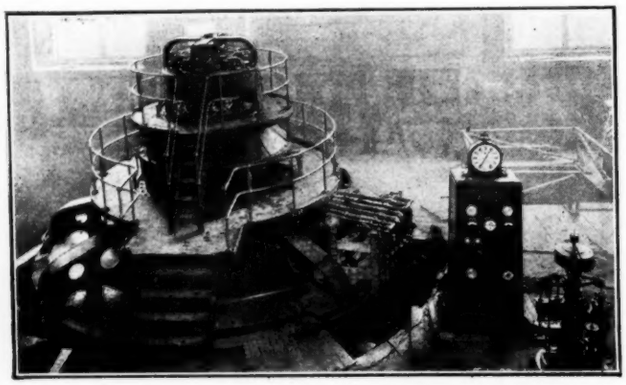


FIG. 12. ACRES CONTROL STAND, GENERATOR AND GOVERNOR

motors for the governor pumps are of the slip-ring type and operate continuously, the water not required for the governors being bypassed through a needle valve from the pressure main to the return main.

Owing to the great amount of power developed in each unit, it is of vital importance that the operator have complete knowledge of all the operating conditions and positive control of the unit at all times. To accomplish these results, a special control stand was built by the Cramp company from designs suggested by H. G. Acres, chief hydraulic engineer of the Hydro-Electric Power Commission of Ontario. This control stand is shown in Fig. 12 and is located on the main generator operating deck. It contains gages showing the operating conditions of the main turbine and principal auxiliaries; namely, gate-opening indicator, pressure on governor system, flow meter for lignum-vitæ bearing, turbine-casing pressure, draft-tube vacuum, air pressure for brakes, electrical meters and telegraph and telephone communicating with the main control room.

The tachometer for indicating the speed of the unit is located on the governor stand, as are also the necessary controls for starting the turbine, adjusting load, shutting down the unit or changing over to hand control; whereas on the control stand are located the air control for the brakes and a control lever for opening and closing the main Johnson valve. It is therefore seen that a single operator from one position can readily observe the performance of the unit and principal auxiliaries and can also start up the unit, adjust the load, change over to hand control or shut down as required in regular operation, and in case of an emergency can also close the main Johnson valve if the turbine gates should become blocked.

In conclusion the writer would like to state that in the design of these units, draft tubes and water passages, the manufacturers were given the fullest co-operation of the engineers of the Hydro-Electric Power Commission with the object in view of obtaining the best possible results in mechanical operation, control, speed regulation and efficiency developed.

Fuses and Conductors Easily Selected for Motor Groups

By William Sheriff Jones

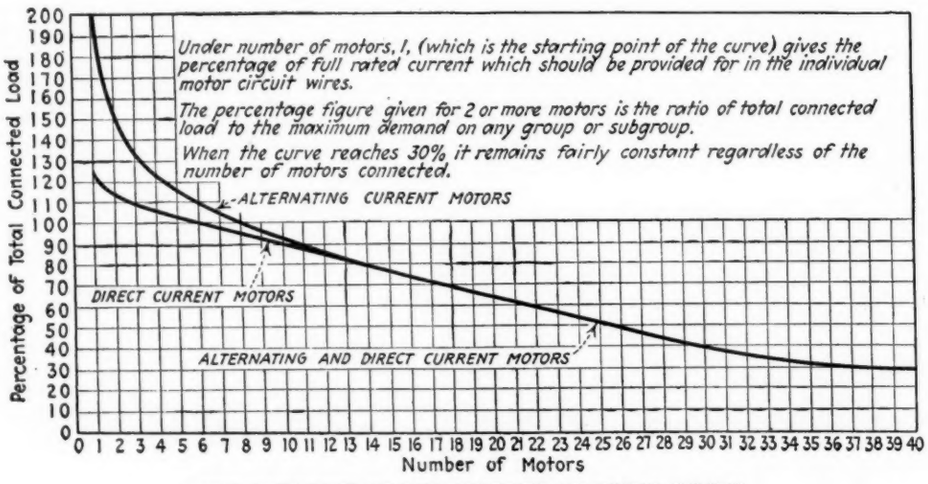
In determining the sizes of wires and cables for motor circuits, the required starting current and the overload capacity of the fuses, which will protect the circuit, should be the deciding factors. The following tabulation gives the ratings and time-element overload capacities of standard commercial fuses:

Rated Capacity (Amperes)	Time Required to Fuse at 150 Per Cent of Rated Capacity
0–30	60 seconds
31–60	2 minutes
61–100	4 minutes
101–200	6 minutes
201–400	12 minutes
401–600	15 minutes

Standard fuses will carry 10 per cent overload for an indefinite length of time. They will fuse at 25 per cent overload, but will carry 50 per cent overload for the time indicated before fusing.

For direct-current motors the circuit to the individual motor should provide a current-carrying capacity at least 25 per cent in excess of that at which the motor is rated to operate normally. For alternating-current motors the circuit should provide from 50 to 150 per cent excess, the lower values applying to synchronous and wound-rotor motors and the higher values to those of the squirrel-cage type.

Where a number of motors are grouped on a single feeder or main, the current-carrying capacity of the feeder or main per horsepower connected becomes considerably reduced as the number of motors increases. The size of the feeder wires required for a group of motors is indicated by the accompanying diversity-factor curve. This curve shows the maximum average demand on the feeder in percentage of the sum of the rated full-load currents of the motors connected to it.


Under number of motors, 1, (which is the starting point of the curve) gives the percentage of full rated current which should be provided for in the individual motor circuit wires.
The percentage figure given for 2 or more motors is the ratio of total connected load to the maximum demand on any group or subgroup.
When the curve reaches 30% it remains fairly constant regardless of the number of motors connected.
ALTERNATING CURRENT MOTORS
DIRECT CURRENT MOTORS
ALTERNATING AND DIRECT CURRENT MOTORS
Percentage of Total Connected Load
200 190 180 170 160 150 140 130 120 110 100 90 80 70 60 50 40 30 20 10 0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40
Number of Motors

FUSE CAPACITY IN PERCENTAGE OF TOTAL LOADS

Steam Traps—Their Selection, Installation and Upkeep

By E. SMILEY*

A TRAP may be defined as an automatically variable opening for the discharge of condensation. This must operate in such a manner as to remove the condensation without any appreciable loss of steam. The discharge may be controlled by varying the amount of the opening or by having the opening intermittent. The corresponding classifications of traps are continuous and intermittent. Traps may also be classified as return and non-return.

Return traps, as the name implies, return the condensation directly to be generated into steam. These are particularly adapted to low-pressure heating systems and other places where condensation on a large scale forms part of the process. In sugar factories and salt works, return traps are used extensively because the water evaporated from the solutions can also be used to good advantage as boiler feed. Non-return traps do not provide for the return of the condensation to the steam-generating unit. Another way of classifying traps is by the mechanism for removing the condensation. On this basis the various types are: Bucket, non-attached float, float, tilt, thermostatic, pot, etc.

A practical knowledge of the selection, installation and upkeep of steam traps is highly prized by the engineer who values his reputation as the guardian of the coal pile. It matters little that the larger equipment is of good design and well operated if there are small steam leaks all over the plant. This article, by a practical expert on traps, gives many valuable pointers, one of the most important of which is the method of distinguishing the actual leakage of steam from the vapor formed by re-evaporation.

The two main factors to be considered in selecting traps are (1) the steam pressure at the trap and (2) the amount of condensation to be handled. It should be carefully noted that the steam pressure desired is that at the trap, not the boiler pressure or the pressure at some point near the trap. Where the pressure varies, the trap should have the correct opening for the higher pressure, yet it should have ample capacity to operate on the lower pressure.

It is foolish to buy a trap by the pipe size with no thought as to the quantity of water to be handled. Capacity tests of various makes of inch-and-a-quarter traps at the Naval Engineering Experiment Station, with 250 lb. pressure, showed a variation in capacity from 2,300 to 26,000 lb. per hour. In other words, some of the traps had over ten times the capacity of others with the same size of pipe connection. The discharge under a given pressure is determined, not by the pipe size, but by the area of the orifice, which is quite a different matter. When steam is on the line or piece of apparatus, the simplest way to find the amount of condensation is to measure it directly, using a graduated measure or bucket and scales.

In general, it may be stated that unless air and water are thoroughly removed, they rest on the bottom of the pipe or cylinder and (even though only a small quantity remains) greatly decrease the effective radiating surface. The best practice is to put a separate trap on each steam coil, drying cylinder, steam main, steam separator, etc.

Where the apparatus to be trapped is not yet installed, it is important to make a reliable estimate of the amount of condensation to be handled. The following data may be useful in this connection:

For ordinary radiators and direct-heating coils using low-pressure steam, the condensation is about $\frac{3}{10}$ lb. per square foot per hour. For indirect-heating coils with forced draft, 2 lb. per square foot per hour is a safe figure under ordinary conditions. However, the front of the stack may run 50 per cent higher than this average. To allow for unforeseen conditions, it is good practice to provide sufficient trap capacity to handle an average of 3 lb. and 5 lb. at the front of the stack. Table I shows the steam that will be condensed per hour by a linear foot of piping (square foot for flat surface or pipe over 12 in.) when bare and exposed to air at 75 deg. F. Suppose it is desired to know the condensation in 80 ft. of 4-in. pipe at 150 lb. gage pressure. The table gives 1.26 lb. per foot. Therefore the total hourly condensation is $1.26 \times 80 = 101$ lb. For flat surfaces or curved surfaces of greater diameter than a 12-in. pipe the last line of the table is used in connection with the square feet of bare surface. For well-insulated surfaces the condensation will ordinarily amount to only 20 per cent of that given by the table.

For steam mains and risers it is customary to multiply the estimated condensation by 4, to allow for flood conditions. The condensation for steam coils or other surfaces submerged in water will depend upon the initial and final temperature of the water surrounding the coil and the metal of which the coil is composed.

Table II can be consulted when selecting traps to handle the condensation from such apparatus. This table may be used in connection with hot-water heaters, feed-water heaters, steam-jacketed kettles, glue pots, submerged boiling coils, etc., and represents the condensation under the average operating conditions with the surfaces somewhat fouled. Where the submerged surface of the metal is clean and will remain so, multiply the table by 2. With liquids containing 10 to 25 per cent of solid matter the condensation is about 25 per cent less than that in the table. In the case of thick viscous liquids the amount of condensation ranges from one-fifth to one-half that shown in the table.

On the drying rolls of paper machines using medium-pressure steam and running heavy paper or felt, there will be about 3 lb. of condensation per hour per square

*E. Smiley and Company, Engineers.

foot of heating surface on the wet end of the machine after the rolls are once heated. This will decrease to about 2 lb. on the dry end. When the paper machine is started with the rolls cold, the condensation on the wet end is often as high as 5 lb. for the foregoing condition. The elimination of air from the dryer rolls of such machines calls for a larger trap than the condensation would indicate.

Low-pressure and exhaust steam, especially that generated from surface water containing much organic matter, is likely to have troublesome non-condensable gases usually called "air." A very simple test for the

TABLE I—STEAM CONDENSED PER RUNNING FOOT OF BARE PIPE EXPOSED TO AIR AT 75 DEG. F., POUNDS PER HOUR

Pipe Size, In.	Gage Pressure, Lb. 5	50	100	150	200
	Temperature Difference, Deg. F. 153	223	263	291	313
1	0.16	0.28	0.36	0.43	0.50
1½	0.21	0.38	0.41	0.59	0.68
2	0.25	0.45	0.59	0.71	0.81
3	0.36	0.63	0.84	1.00	1.16
4	0.45	0.79	1.05	1.26	1.45
5	0.54	0.97	1.28	1.54	1.78
6	0.64	1.14	1.51	1.82	2.09
8	0.82	1.47	1.94	2.35	2.69
10	1.01	1.80	2.39	2.89	3.30
12	1.19	2.13	2.82	3.40	3.91
Flat surface, or larger pipe, per sq.ft....	0.34	0.62	0.82	0.99	1.14

presence of excessive "air" in the steam lines is to run a length of pipe or rubber tubing well beneath the surface of the water in a bucket and watch the bubbles. Any gases that are not entrained in the water will be indicated by the bubbles.

When a trap is attached to a piece of apparatus using superheated steam, care must be taken that the steam is not superheated when it reaches the trap, otherwise the latter will not operate. Drips on superheated lines are necessary to handle such conditions as the priming of the boilers. In such cases the connection to the trap must have enough radiating surface to insure condensation for the trap to work on. Traps of the continuous-discharge type and many of the bucket traps require a certain amount of water in the body. If superheated steam gets into the trap, it re-evaporates the water and causes leakage and wiredrawing. On

TABLE II—CONDENSATION FROM STEAM COILS SUBMERGED IN WATER*

Difference Between Temperature of Steam And Average Temperature of Water, Deg. F.	Approximate Weight of Steam Condensed Per Square Foot of Submerged Surface, Pounds Per Hour. Wrought Iron Coils	Brass Coils	Copper Coils
50	5	8	9
100	16	25	29
150	33	50	57
200	55	82	93
250	81	122	138
300	110	165	190

* Table from American Radiator Co.

account of the tendency to wiredrawing with superheated steam and with apparatus operating at high pressure with but little condensation, a special metal should be provided in the valve and seat.

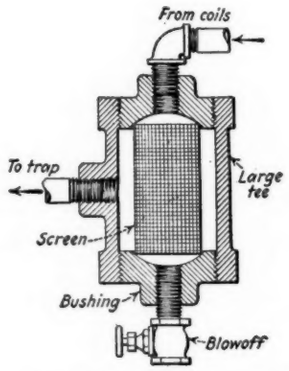
The question of scale, dirt and oil is a troublesome one with most traps when there is an excess of any one of these in the line. Scale is to be expected in all systems, especially on intermittently heated surfaces where air can get at the pipe interiors when the pipes are cold and drained. Dirt and sediment are usually encountered when the piping is first made up, and they may also come from the action of boiler compounds or they may be picked up by the steam in leaky process piping.

The best way to avoid trouble from dirt, scale or sediment is to use a strainer like that shown in the figure. This is easily made up from pipe fittings and a strip of heavy brass or copper wire screen, made in the form of a cylinder so that it will fit in the recesses of the bushings. Of course such a strainer requires occasional cleaning, but that is a simple matter compared with cleaning a trap.

Many traps are provided with blow-off pet-cocks or valves so that the traps can be blown down and cleaned out periodically. A simpler but less efficient remedy is a dirt leg in the piping in front of the trap. This can be made up with a tee, long nipple and cap. Excessive oil in the line can usually be traced to a faulty oil separator in the system.

The only way to be absolutely sure that a trap is clean is by an examination of the interior. In fact, it is good practice to inspect traps at least once or twice a year. Some large plants have trap inspectors who continually make their rounds.

Coming to the outlet side of the trap, we must consider the question of back pressure, which may be defined as the pressure on the discharge side of the valve seat, measured above atmosphere, against which the trap must work. It is caused by: (a) A head of water; (b) a discharge into a receiver under pressure; (c) an obstruction in the discharge line due to sediment, scale or valves; (d) pipe friction. Several of these causes may be present at the same time.



EASILY CONSTRUCTED STRAINER

Although 1 lb. pressure at the trap will support 2.4 ft. of water at 212 deg., it is well to figure back pressure at the rate of about one pound for every two feet of head. This makes a reasonable allowance for friction in the pipe and fittings.

In general, the back pressure should be deducted from the pressure on the trap to get the differential or effective pressure on the trap, which in turn governs the size of orifice to be used or the capacity that can be obtained from an orifice of fixed area.

Not all traps are capable of lifting the discharge water. Some non-return traps of the continuous type should not be used for lifting with a trap pressure less than 3 lb. gage. Above that this type of trap can raise about 2 ft. for every additional pound pressure at the trap.

In the case of return traps, the capacity is in proportion to the head above the boiler-water line. For example, if the trap chamber, under given conditions of pressure and piping, is located 4 ft. above the boiler-water line, a change of 1 ft. in the level will make a change of 25 per cent in its capacity.

In order to decrease the head due to friction, return traps should be placed near the boiler. However, if the location does not permit this, the difficulty may be largely offset by using larger discharge lines than the trap sizes call for and by avoiding the use of closed valves in these lines. It is well also to keep in mind that where a return trap is somewhat under size and the pressure of inlet will permit, a higher elevation of the trap will increase the capacity.

Return traps may be located below the boiler to be fed, so that low-pressure returns will flow into them,

provided that steam at a higher pressure than that in the boiler to be fed is available from some other source. This is required in order to discharge the trap to the boiler. However, it may be arranged to have the receiving trap discharge into an elevated tank fitted with a steam connection and a valve connected to and operated by the trap. This is so laid out that when the trap is discharging there is no pressure on the elevated tank, and while the trap is filling the elevated tank is under steam pressure and discharging the water into the boiler.

TABLE III—VOLUME OF RE-EVAPORATION FROM WATER DISCHARGED BY A CONTINUOUS (FLOAT) STEAM TRAP TO THE ATMOSPHERE.

Gage Pressure At Trap	Per Cent Re-evaporation	Ratio By Volume Of Vapor Discharged To Discharged Water
5	1.6	26
10	2.9	47
20	4.9	82
40	7.8	136
50	9.0	158
100	13.3	246
150	16.3	313
200	18.7	370
250	20.7	418

It is also important to know whether the discharge from non-return traps is into a vacuum return, because every 2 in. of vacuum is equivalent to an additional pound pressure on the trap. For example, with 1 lb. pressure at the trap, and 10 in. of vacuum, the pressure differential at the valve would be 6 lb., and an orifice with the proper area for this pressure should be used. Furthermore, certain traps have special orifices for vacuum, so that in selecting a trap it is well to include the data on a vacuum return.

SOME DON'TS

1. **Don't buy a trap that is impossible for the average man to repair. (Interchangeable valves and seats are a commendable feature).**
2. **Don't allow traps to be placed in inaccessible locations.**
3. **Don't fail to have traps so connected that a quick and accurate test may be had for leakage.**
4. **Finally, don't fail to keep a systematic record of trap inspection.**

High-pressure traps should not in general discharge into sealed return pipes under a low pressure, as the breaking into steam (or flashing as it is called) of the high-temperature water upon release will cause snapping and water hammer in the pipes. It is usually best for all traps, when the condensation is to be saved, to discharge into a vented receiver above the water line. This prevents any backing up or water hammer, and also gives an opportunity to detect severe steam leakage by an inspection of the outlet to the vent pipe leading from the trap.

Many a perfectly good trap has been scrapped and many a good new one never bought on the ground that it leaked steam. For the benefit of those who may not yet be familiar with re-evaporation, it should be pointed out that when a trap is connected to a steam line or apparatus containing steam at 100 lb. gage pressure, the temperature of the water in the trap, just before it is discharged, is about 338 deg. At atmospheric pressure water boils at 212 deg., so that when water whose temperature is over 212 deg. is forced out through the trap outlet into a space under atmospheric pressure, it will naturally boil or evaporate until its temperature drops below the boiling point—that is, 212 deg. This boiling or re-evaporation causes the vapor, which is sometimes mistaken for steam leakage.

Occasionally a person, while admitting the foregoing, will claim that there seems to be more vapor or steam rising from the discharge than would be caused even by re-evaporation. This feeling is generally due to the fact that few people appreciate how great a volume of vapor may be caused by the re-evaporation of a comparatively small amount of liquid. Table III has been compiled to enable the reader to form a picture of the volume of vapor given up by a certain amount of water under certain conditions.

It will be seen that this is much greater than one would ordinarily expect without actually figuring the problem. In fact, with continuous-flow traps re-evaporation indicates a normal and efficient trap discharging the condensation as quickly as it receives it. If this discharge is returned to a feed-water heater, more heat is saved than if the discharge from the trap were at a lower temperature.

The fairest and surest way of determining whether a trap leaks steam, is the simple little test outlined below. Run a length of pipe from the discharge well below the surface of water in a bucket about half full of water. In case the discharge is more than 300 lb. per hour, a more accurate test can be made with a larger container, such as a barrel about half full of water. The test is started with cold water below the temperature of the room and continued until the water is heated to a temperature about as much above that of the room as it started below. This tends to equalize any error due to radiation, etc. Where the rate of flow is less than 300 lb. per hour, the weighing should be carried out to ounces, as the flow is taken for only a short time. With a greater flow the usual platform scales are accurate enough.

The data required are as follows: Initial temperature and weight of the water in the pail; final temperature and weight of the water. The gain in weight represents the hot water or hot water and leaking steam that has come from the trap. By use of the steam table it is possible to figure how much heat the trap exhaust, if all water, could give up to the water in the pail. If the actual heat gained by the water in the pail is more than this, it shows that there is some leakage of steam. The following is an example of how this is figured:

Weight of container and cold water	14 lb. 5 oz
Weight of container alone	4 lb. 5 oz
Net weight of cold water	10 lb.
Temperature at end, deg. F	90
Temperature at start, deg. F	50
Increase in temperature, deg. F	40
Heat added to cold water = 10 lb. x 40 deg., B.t.u.	400
Steam pressure (gage at trap), lb.	100
Heat in 1 lb. of discharged water at steam temperature, B.t.u.	308.8
Heat in 1 lb. of discharged water at end of test (90-32) B.t.u.	58
Heat given up by 1 lb. of discharge water, B.t.u.	250.8
Weight of water required to bring in heat actually found (400 ÷ 250.8), lb.	1.59
Weight of water actually discharged, lb.	1.62

Report: Trap does not leak steam, as the weight of water actually found is greater than the weight required.

For plants where the traps are subject to excessive vibration the following are desirable features of construction and installation:

1. Strong construction with properly designed gasket joints which may be kept tight without difficulty.

2. Inlet and outlet connections in the body of the trap.
3. Outlet valve located above the bottom of trap where it is not likely to become clogged.
4. An accessible inlet strainer.
5. Power valve-operating mechanism.
6. Gage glass or other device for easily determining whether trap is operating properly.
7. Connection to the lines with unions and with by-passes around most traps.

A general rule on the location of a trap is to keep it below the lowest water level in the unit it is draining. This keeps the discharge line free of condensation when the pipes get cold on intermittent service, and thereby decreases the possibility of water hammer.

Traps that are outside and likely to be injured by freezing should be insulated and located as near the heating unit as possible. Their discharge lines should also be automatically drained in case they rise above the lowest water level of the trap. This prevents the condensation from flowing back into the traps when the pressure is taken off. It is of primary importance that they be accessible for periodic inspection. An inspection of many a trap would reveal a loss of steam that would quickly repay the repair or replacement of the present trap. At 100 lb. pressure, 14,000 lb. of steam will waste to the atmosphere in six days of nine hours each through a leak having an area of $\frac{1}{25}$ sq.in. With steam at 50 cents a thousand pounds, this would mean $7 a week loss, so that it would not take many weeks to pay for a whole new trap.

A change of engineers in a large manufacturing plant was attended by a saving of $200 per month in the coal bill, due to overhauling a family of 50 steam traps. This department alone was the undoing of the former engineer, who unfortunately believed the tales concocted by his repair crew.

Condenser-Water Supply for CO_2 Refrigerating Machines

By A. E. Kimberly

Carbonic anhydride refrigeration is usually understood to require cold condensing water. While water of low temperature is undoubtedly highly desirable, as pointed out by Prof. Macintire, in *Power*, Jan. 3, 1922, partial liquefaction of the compressed gas can be secured with cooling water at temperatures as high as 80 or 90 deg. F., although, of course, at reduced efficiency.

To develop a supply of ground (well) water, the author was recently retained by a hotel company operating two carbon-dioxide machines, one of 25 and one of 45 tons daily capacity. These machines, driven by Corliss engines, furnish the refrigeration required by ice and ice-cream making, food chilling and air cooling for the dining rooms, assembly halls, lobbies and engine room.

For about seven months of the year municipal water, a filtered supply from a river, was used over the condenser and under the somewhat light refrigeration load of these seven colder months, had a temperature sufficiently low to permit refrigeration at pressures not to exceed 60 atm. (900 lb.).

In the five summer months, the temperature of the municipal water supply ranges from 70 to 80 deg. F. and with the increased refrigeration load incident to maximum demand for air cooling, condensing was difficult and expensive owing to the high pressure of the gas. In addition its near approach to the critical temperature (88.4 deg. F.) caused much loss through imperfect liquefaction, resulting in reduced refrigeration efficiency as well as reducing the output of the plant when the demand was greatest.

Under instructions to inquire into the feasibility of developing a sufficient supply of relatively cold water for the condensers, it was necessary first to determine the approximate volume of water required and to obtain data as to the quality, temperature and volume of water available from underground sources in the vicinity of the hotel.

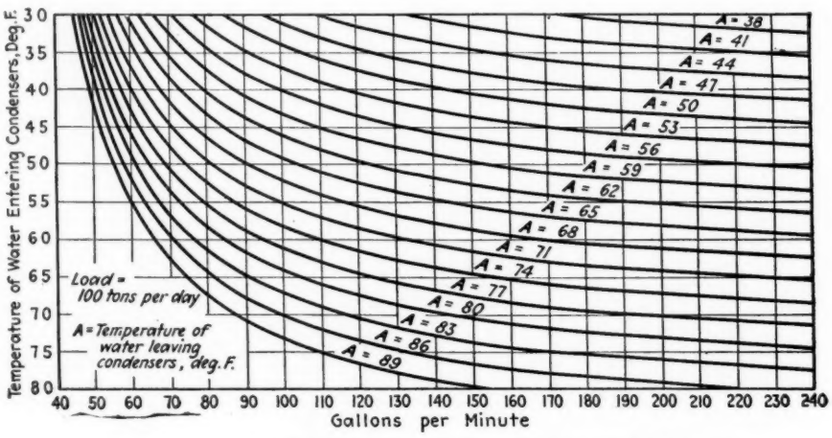


FIG. 1. COOLING WATER REQUIREMENTS FOR CO_2 SYSTEMS

In directing inquiry into the question of quantity, a matter on which it appears but little has been written with respect to refrigerant condensing, a formula was developed converting the well-known relation 200 B.t.u. per min. per ton of refrigeration per day into volume of water. The volume of condensing water depends, of course, upon the temperature of the condensing water, the pressure of the gas and the load, and may be expressed by the formula:

$$x = \frac{200\,b\,C}{8.34\,(a - y)}$$

where x = Gallons of cooling water per minute;
y = Temperature of water entering condensers;
a = Temperature of discharge water;
b = Refrigeration load in tons per 24 hours;
C = A coefficient.

The coefficient C takes into account the heat produced in the compression of the gas. This heat must, of course, be abstracted by the cooling water in addition to the heat of vaporization of the refrigerant. The value of C is variable and dependent upon the pressures maintained in the condenser and in the brine coils, the greater the difference in these pressures the higher its value. Assuming that the pressure in the condenser varies from 800 to 1,000 lb. absolute and that the pressure in the brine coils varies from 300 to 500 lb.

absolute, the value of C will vary from 1.11 to 1.36. as shown by the following:

Cond. pres.....	800	800	800	965	965	965	1,000	1,000	1,000
Coil pres.....	300	400	500	300	400	500	300	400	500
Coeff. C.....	1.22	1.16	1.11	1.31	1.23	1.17	1.36	1.29	1.22

Since these ranges of pressure correspond to usual operating conditions, an average value of C of 1.25 is sufficiently accurate for all practical purposes.

For a given refrigeration load it is essential that at least sufficient water be supplied to the condensers to

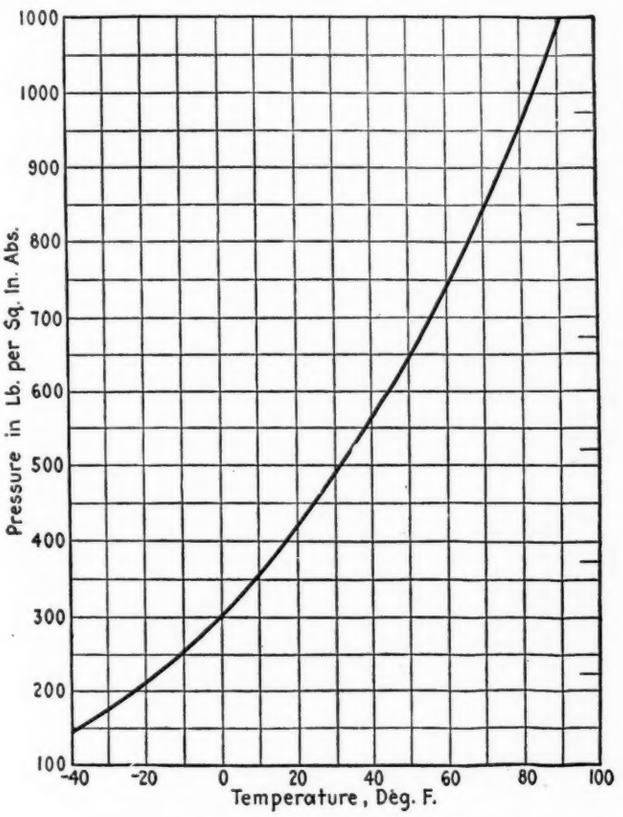


FIG. 2. RELATION OF PRESSURE AND TEMPERATURE OF CARBON DIOXIDE

abstract the aforementioned quantities of heat, maintaining at the same time the particular pressures desired on the high and low sides of the compressor. In this connection, it should be noted that the temperature of the condensed gas is usually from 8 to 12 deg. F. higher than that of the condenser tail water, and due allowance should be made when computing the volume of water required.

To simplify these computations, the chart in Fig. 1 is presented showing the relation between the variables in the foregoing equation with the exception of the coefficient C. Hence, the values obtained by the use of this chart should be multiplied by 1.25. To illustrate the application of the chart, in connection with the chart of carbonic anhydride properties shown in Fig. 2, assume a load of 50 tons per day, a temperature of condensing water of 54 deg. F., and a maximum pressure of 975 lb. From the chart of properties of carbonic anhydride, Fig. 2, the temperature at 975 lb. is 82 deg. F., and allowing 8 deg. difference, the discharge water temperature is 74 deg. F. The quantity of cooling water as shown by the chart in Fig. 1, is 60 gal. per min. and multiplying by the coefficient C. 75 gal. per min. for a 50-ton load. Similarly, for other temperatures and pressures the volume of water can readily be determined for any refrigeration load.

A decision was finally reached that the limestone beneath the hotel would furnish an ample supply of water of relatively low temperature, satisfactory at least for condensing purposes. Arrangements were accordingly made for well drilling, and an 8-inch hole was drilled to a depth of 176 ft., entering the rock for a distance of 44 ft. The well was successful and supplies about 135 gal. of water per min. at 54 to 55 deg. F.

Welding Underground Steam Mains*

Although welded pipe joints for underground steam construction are not the general rule, several district heating companies have used them and predict a coming field for them. The advantages claimed for welded joints over the ordinary screwed joints are as follows: One welded joint takes the place of two screwed connections. The welded joint, properly made and tested, should require no maintenance during the life of the pipe. In laying larger sizes of pipe with screwed joints, difficulty is often experienced in lining up the pipe and there is always the danger of getting the threads crossed. These difficulties are not encountered in welding. The application of sectional covering gives less trouble and is neater and more compact on welded pipe.

In making butt welds, one of the most important precautions is to space the pipe properly so that there will be no failure at any point to penetrate thoroughly the full thickness of metal with the weld. Far better to have the weld protrude beyond the inner surface than to lack complete penetration. To insure the best weld, the pipe should be beveled carefully at an angle of from 30 to 45 deg. with the horizontal axis, leaving a square end of about one-quarter the full thickness of the pipe. After spot-welding in four to six places around the pipe, sufficiently strong to maintain its position, the inner bead should be made with care in connecting the inner circumference, following with the second bead or reinforcing weld. The thickness of the reinforcing weld depends somewhat on the pressure to be carried in the line, but it seems rather inconsistent to build up the weld much thicker than the pipe itself.

Whenever practical, welding should be done on the top surface. Rollers may be used and the pipe turned by helpers with chain tongs. This will facilitate the welder's work. When an expansion joint or a valve is to be installed, it is necessary for the welder to work around the pipe as it lies in the trench.

An overhead weld must be made in welding the bottom of the pipe, and this may be accomplished successfully when welding electrically by increasing the current to drive the metal in. For turns in the line, pipe bends anchored at both ends are used.

The general method of testing welded pipe is by hydraulic pressure. It has been found that an imperfect weld will show a water leak under little pressure. After the hydrostatic test it is advisable to turn steam under pressure into the line to test for expansion.

The cost of electrically welding pipe is dependent largely on the conditions under which the work must be done, but a price of from 30 to 50 cents per lineal inch is a fair average, including labor for handling pipe.

*Excerpts from report of the Underground Construction Committee, National District Heating Association, Cedar Point, Ohio, June 20-23, 1922.

Locating Faults in Direct-Current Armatures—Short-Circuits in Groups of Coils*

Effects of a Short-Circuit in a Group of Coils in Armature Windings—Methods of Locating the Faults

By B. A. BRIGGS

SHORT-CIRCUITS may occur in armature windings so as to involve only a single coil as discussed in previous articles, or these defects may include a number of coils. The latter will be the subject for consideration in this article. Where only one or two coils are short-circuited in a winding, the only effect noticeable is excessive heating of the coils, which may cause the insulation to smoke in a very short period of operation, or, if not discovered in time, will cause complete destruction of the insulation on the defective coil and injury to other coils in proximity to the faulty one.

Where a number of coils are involved in the fault, not only will the coils be heated, but the machine will act as if it were heavily overloaded. When starting, if the machine is a motor, the armature will revolve slowly with a jerky motion and will draw a heavy current from the line, which will cause the starting resistance to become abnormally hot. If an attempt is made to cut the resistance out of circuit, the protective device will open. With a generator the prime mover will give every indication that it is starting the machine under a heavy overload. However, with either a motor or generator a section of the winding will increase to an abnormal temperature, and to those familiar with such symptoms there can be little doubt as to the cause of the trouble.

In Fig. 1, if the winding is moving in the direction of arrow *K* and the field poles are above the winding, a voltage will be generated in the coils in the direction of the arrows shown on the coils. Assume that a short-circuit occurs between coils 8 and 13 at *X;* this will create a close circuit for coils 9, 10, 11 and 12, as may be seen by following out the circuit. This circuit starts at *X* and is down the top side of coil 13 to lead *c*, then through coils 12, 11, 10 and 9, then up lead *d* into the lower half of coil 8 and to *X*, thus completing the circuit. Part or all of coils 8 and 13 may be involved in the fault, depending upon how the defect is made between the two coils. If the contact resistance between coils 8 and 13 at *X* is low, a heavy current will circulate in the coils involved in the fault, even at low speeds, which will produce a strong retarding effect in the armature and will cause the machine to act as if overloaded.

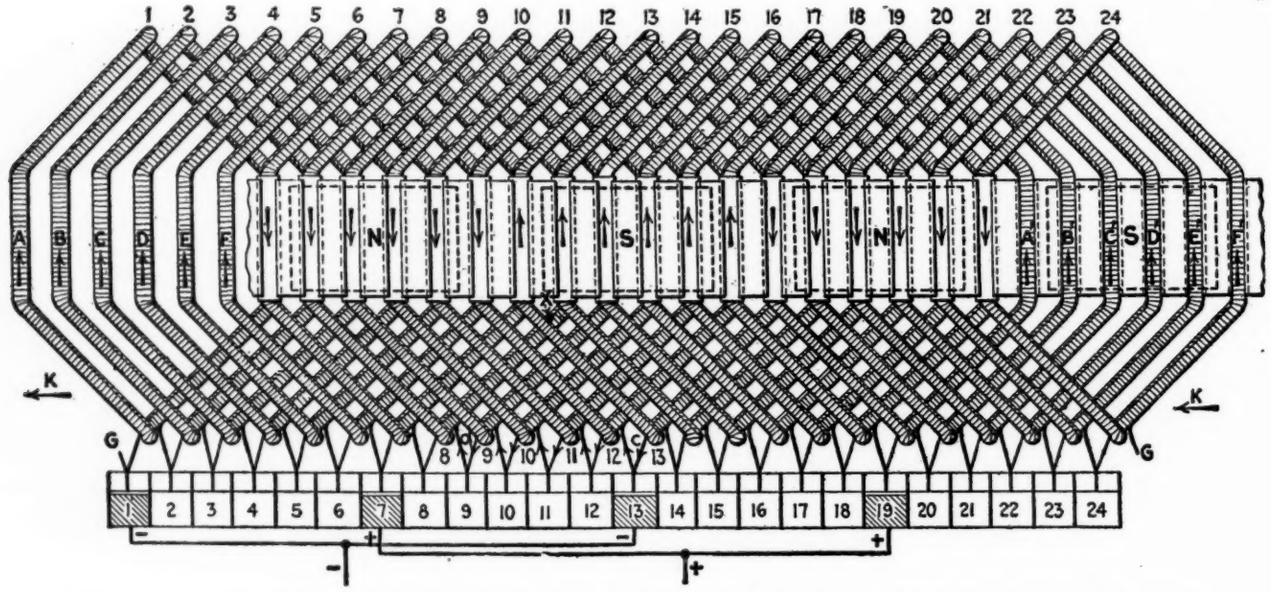


FIG. 1. PARALLEL WINDING SHOWING HOW CURRENT FLOWS IN A GROUP OF SHORT-CIRCUITED COILS

When a current is passed through the parallel winding, Fig. 1, to locate the defect with a low-reading voltmeter, as indicated in Fig. 2, if a metal-to-metal contact is made between coils 8 and 13 at *X*, little or no current will flow through coils 9, 10, 11, and 12. One circuit through the winding is from the brush on segment 7 through coils 6, 5, etc., to coil 1 and lead *G*, then through coils, 24, 23, etc., to coil 19 and the negative brush on segment 19. The other circuit is through coils 7 and 8 to *X*, then up around through coil 13 to segment 14, through coils 14, 15, etc., to coil 18 and segment 19 and the negative side of the line. If the contact resistance is comparatively high at *X*, considerable current may flow in coils 9, 10, 11 and 12, so that when readings are taken on these coils with the milli-

*All rights reserved.

voltmeter the deflection may amount to a considerable part of normal.

Assume that the only defect in the winding is that at *X*, then if readings on the millivoltmeter are taken on, say, segments 18 and 17, 17 and 16, 16 and 15, 15 and 14, they will be normal, say three-quarters the way across the scale. Now when a reading is taken on segments 14 and 13, the deflection obtained will depend upon how much of coil 13 is involved in the fault. If only a small part is at fault, then the throw of the needle on the scale will be near normal, where if nearly all the coil is included in the fault the reading will be low. Low readings will also be obtained on coils 12, 11, 10 and 9, showing that this group of coils is short-circuited. Finding the actual defect involves lifting the leads from the segments where the low readings are obtained and then raising the top side of the coils, to which the leads connect, until the fault is found.

With the series winding a short-circuit between coils or leads will cut out two groups of coils. For example, in Fig. 3, assume that there is a short-circuit between the leads of coils 10 and 13 at *X*. With the armature moving in the direction of arrows *K*, voltage will be generated in the coils as indicated by the arrows on the coils. The short-circuit at *X* provides a close circuit in the winding from *X* through coils 13 and 25 to *l'* and *l* then through coils 12, 24, 11, 23 and 10 back to *X*, thus completing the closed circuit. The voltage generated in the coils will cause a heavy current to circu-

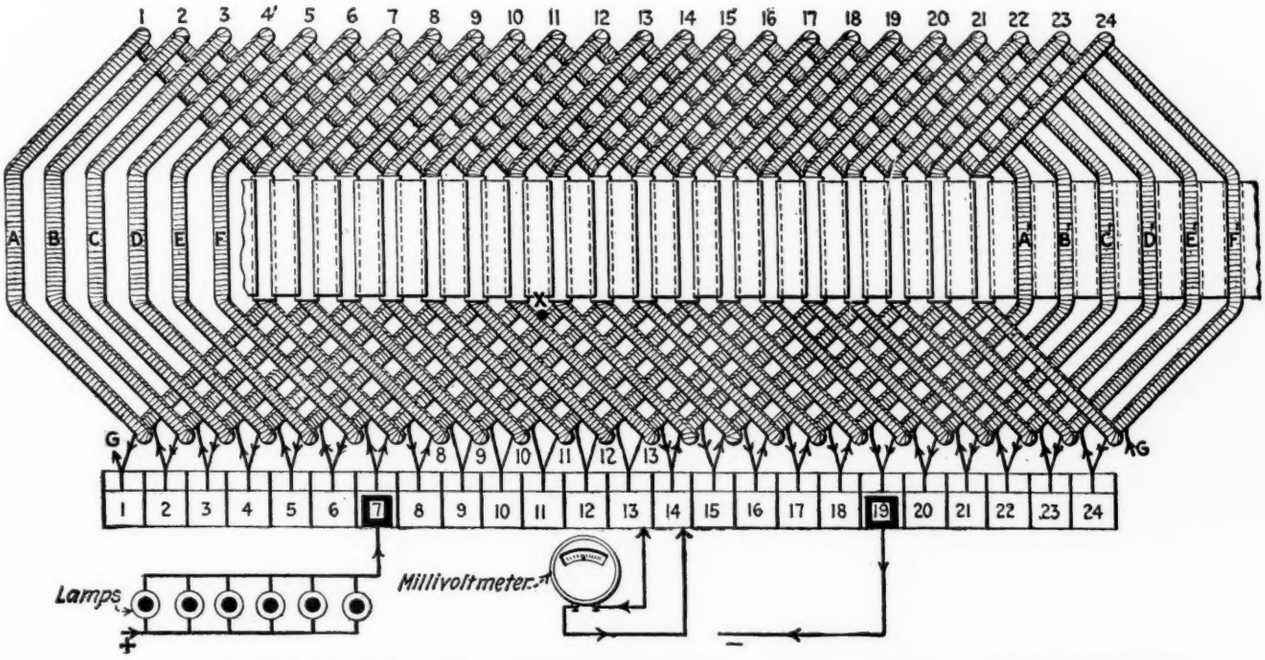


FIG. 2. METHOD OF TESTING FOR A GROUP OF SHORT-CIRCUITED COILS IN A PARALLEL WINDING

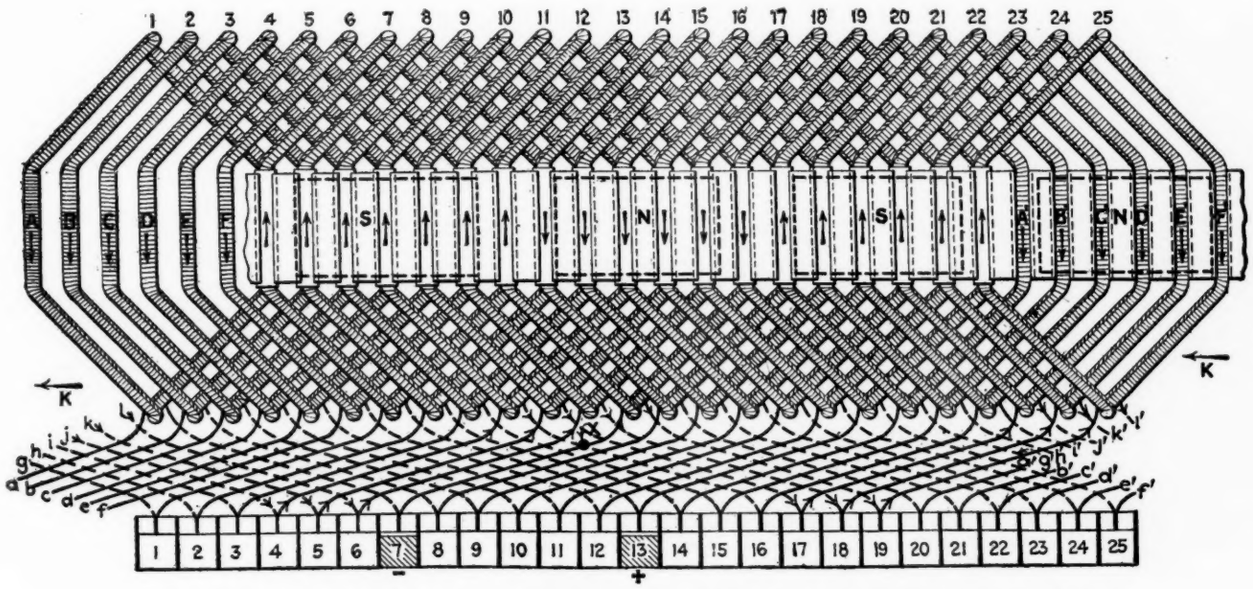


FIG. 3. SERIES WINDING SHOWING EFFECTS OF A SHORT-CIRCUIT BETWEEN COIL LEAD

late and produce the conditions of an overload on the machine as well as heating of the coils, as explained for the parallel winding. It will be noted that instead of one group of coils being short-circuited as in the parallel winding, there are two groups in the series winding, Fig. 3. As it works out in a series winding, there will be one group of coils short-circuited for each pair of poles.

If current is passed through the winding from an outside source, as in Fig. 4, so as to locate the faults with a low-reading voltmeter, there will be a section of the commutator for each pair of poles that will give a low deflection on the instrument. In tracing out the circuit, in Fig. 4, it will be seen that the fault at *X* provides a circuit from segment 7 to segment 16 instead of through coil 13 to segment 19. This cuts coils 13, 25, 12, 24, 11, 23 and 10 and leaves segments 4, 5, 6, 17, 18 and 19 at a low potential so that when readings are taken on these bars the deflection on the meter will be small, thus indicating that two groups of coils are short-circuited in the winding, generally referred to as a short-circuit on the quarter. In a six-pole winding there would be three groups of bars on the commutator from which a low reading would be obtained, and in an eight-pole series winding there would be four groups of segments giving a low-reading when a group of coils is short-circuited in the winding.

After the groups of segments that give the low readings have been located, then the top leads connecting to these bars may be lifted, also from an adjacent bar on each side of those giving the low reading. If the defect is between coil leads, as at *X* in Figs. 3 and 4, the place where the insulation is defective should be easily found. Should the fault be between the coils, it will be a case of lifting the leads from the bars and testing each coil to all the rest.

The foregoing test can be made by lifting, say, the top lead from segment 4 and then counting around the commutator, in the direction of the other lead from coil 10, the pitch of the coil leads on the commutator. In this case the count would be 1 to 13 to the right. Calling No. 4 segment 1 and counting to the right 13 ends on segment 16 to which the other lead of coil 10 connects. Now if both leads are lifted from segment 16, the bottom lead can be tested with a test lamp to the top lead taken from segment No. 4. After it has been ascertained that the two leads of one coil have been located, this coil should be tested to the rest of the coils, to see that it is not short-circuited to some of them. This is done by connecting one lead of the test circuit to one of the coil's leads—for example, the lead lifted from segment 4—and with the other test lead test on the other segment such as 5, 6, 7, 3, etc. If no light is obtained, then lift the top lead from segment 5 and both leads from segment 17, and with one test terminal on the lead from segment 5, test the coil for short-circuits between it and the other coils in the winding.

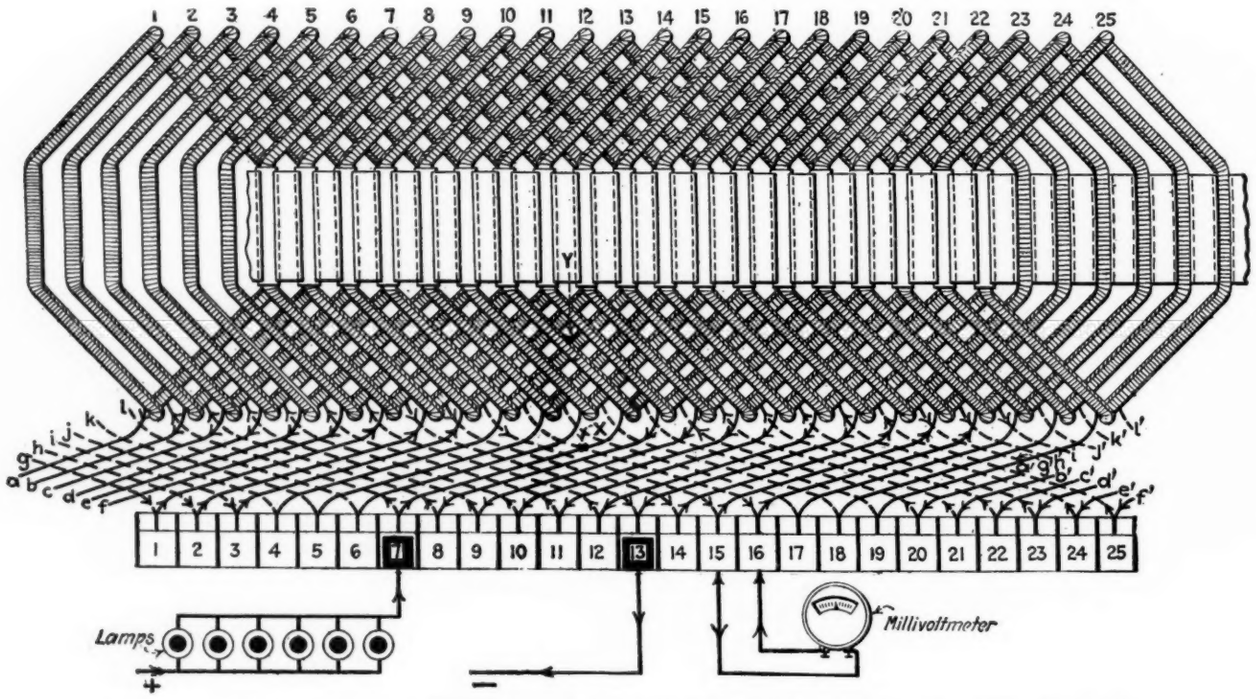


FIG. 4. METHOD OF TESTING FOR A GROUP OF SHORT-CIRCUITED COILS IN A SERIES WINDING

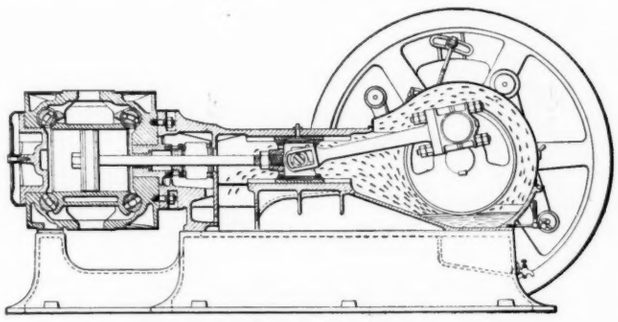
Suppose that instead of the fault being between the leads of coils 10 and 13 at *X*, it is between the coils at *Y*. Then if coil 10 is isolated by disconnecting its leads from commutator bars 4 and 16 and one terminal of a test circuit is connected to one of the coil's leads and the other test terminal is placed on segment 7, the circuit will be completed through coil 10 to *Y* and then through coil 13 to segment 7 and the lamp will light. Of course a light would also be obtained on practically all of the other segments in the commutator; nevertheless, the fact that a light is obtained from coil 10 to the others when its leads are isolated from the commutator, shows it to be one of the defective coils, after which it should be an easy matter to find the actual location of the fault. The next article will deal with the location of open circuits in parallel and series connected windings with a low-reading voltmeter or with a telephone receiver.

The Problem of Lubricating Steam-Engine Bearings

By W. F. OSBORNE
Supervisor Manufacturers' Service, Texas Company

THE bearings of steam engines present a wide variety of conditions with respect to lubrication. There are many factors to be taken into consideration in the selection of a suitable oil, and each must be weighed carefully if satisfactory results are to be secured. Although the influencing factors are the same, the order in which they are investigated depends upon whether the question is one of designing a new bearing to give minimum friction or one of selecting a suitable lubricant for an existing engine. It is in this latter situation that most of the readers of *Power* will be interested, and this phase of the subject will be considered.

The first point to take into account is the method of lubrication. Are the bearings lubricated by a squirt



KEEPING WATER OUT OF ENGINE FRAME

can, drip cup, manifold reservoir, ring oiler, splash system, gravity or force feed? When any one of these methods is used, does it provide perfect or only partial film lubrication?

The first three methods in most instances will not build up and maintain a perfect lubricating film because the small quantity of oil supplied to the bearings is not sufficient to keep the clearance space full, on account of oil leakage from the ends of the bearing. Hence the shaft is not fully supported on a film of oil, and conditions of partial or so-called greasy friction must exist. A lubricant should have a viscosity sufficiently high to induce wedge action and to remain in the bearing as long as possible.

The last four methods of applying the lubricant, if properly arranged, will furnish an ample supply of oil to maintain a perfect oil film, provided the mechanical conditions and the viscosity of the oil are such as to permit it. Generally speaking, the oil should have the lowest viscosity that will form and maintain the lubricating film with a sufficient margin of safety. An unnecessarily high viscosity will increase the frictional losses within the bearing and raise the steam consumption. To many engineers this may seem to be a very small source of loss, but it actually amounts to as much as 50 per cent of the total bearing friction loss when an excessively heavy oil is used. It is a common occurrence to lower bearing temperatures by the use of a low-viscosity oil, and of course the frictional losses are correspondingly reduced.

As was explained in *Power*, April 18, the formation of the oil film is dependent upon a large number of factors—r.p.m., load per square inch, bearing clearances, oil grooves, the quantity of oil supplied, etc.—so that it is quite impossible to make any specific recommendations that will apply to all cases. However, most engines equipped with circulatory systems can be lubricated with an oil of between 150 and 200 seconds Saybolt viscosity at 100 deg. F. When the drip cup or similar appliance is used, it is frequently necessary to raise the viscosity to 250-300 seconds, and sometimes when speeds are low, loads high, or bearing clearances are very large, it is necessary to use viscosities as high as 450-500 seconds.

Because of the low speed at which most steam engines operate, the formation of a perfect oil film is a rather difficult matter, unless all conditions are properly adjusted. On this account the circulatory system is of great advantage in securing best results because it furnishes a large volume of oil to all the bearings.

The oil on a steam engine is subjected to one form of contamination which most other oiling systems do not have to take care of. This is the contamination due to admixture of condensed steam carrying emulsified cylinder oil. When a steam cylinder is properly lubricated the water leaking through the stuffing box will carry emulsified cylinder oil into the crankcase, particularly if a compounded oil is used. This emulsified cylinder oil and water has a great tendency to emulsify the rest of the engine oil in the system. The high viscosity of the cylinder oil also raises the viscosity of the engine oil, and the increased viscosity, together with the content of emulsified animal oil, causes the engine oil to deteriorate rapidly. A greater load is placed on the cleaning apparatus, and the loss of good oil which has been partly emulsified is increased. Special arrangement of filters and separators is frequently required in order to maintain the oil in a proper condition.

To avoid these troubles, many engine builders have equipped their engines with a double stuffing box or bulkhead to prevent this water and oil from getting mixed with the engine oil, as shown in the illustration. The water is drained away and the engine oil remains in much better condition. The bulkhead has the additional advantage of preventing the engine oil from splashing up against the hot cylinder head, which would cause increased evaporation and decomposition of the oil.

When the engine crankcase is not completely inclosed, there is likely to be loss through splashing of oil out onto the floor and through increased evaporation. The warm temperature within the crosshead guide barrel and the crankpit, together with the splashing of the oil, increases the evaporation, and unless the vapor is held within the crankcase, it will float out into the engine room. Sometimes the roof trusses will be covered with

drops of oil that have condensed from the vapor in the engine-room air. If there is no bulkhead to keep the oil off the cylinder head and to keep out the leaking steam, it is sometimes quite difficult to inclose the crankcase completely, on account of the high temperatures that may develop, and it is necessary to have a circulation of air to keep things cool.

When the circulatory system is provided with efficient filters and separators, a good grade of red engine oil can be used very satisfactorily, the viscosity depending upon the individual engine conditions. If the capacity of the system is small, the rate of circulation rapid or the contamination from moisture and other foreign matter large, better results can be secured from the use of a filtered engine oil. The filtered oil will separate more readily from contaminating matter, its rate of decomposition will be lower, and the oil will remain in good condition for a longer period.

Many Engines Not Equipped with Filtering Apparatus

Many of the inclosed crankcase splash-lubricated engines are not equipped with any sort of filtering apparatus, and the oil never gets cleaned at all except when the engine is shut down. Then some of the moisture and sediment will separate, but it is usually stirred up again as soon as the engine is started. An oil for such a system should be as highly filtered and as carefully refined as a steam turbine oil. In fact a turbine oil is frequently used. The viscosity should be the very lowest that can be secured and still have a sufficient margin of safety. The low viscosity permits of the rapid separation of the moisture and the foreign material.

Such engine systems should be completely drained out at intervals and the crankcase washed out with kerosene and finally with a little of the fresh oil. Many of these engines are in use in boiler plants, and quantities of coal dust find their way into the crankcase. The coal dust, together with the condensed water leaking past the stuffing boxes, forms a bad emulsion which quickly destroys the bearings.

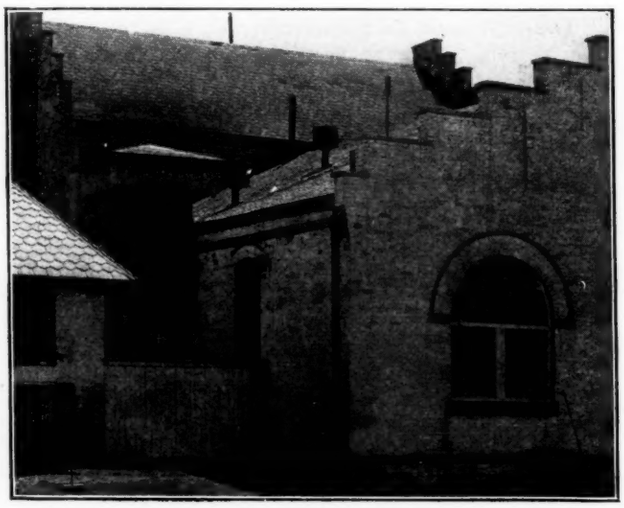
One other factor must be considered and that is the pour test of the oil. In a few instances engines are located out of doors, exposed to low temperatures of winter weather. If one of these engines is shut down occasionally so that the oil is not warmed from the cylinder, the oil in the crankcase will congeal and fail to lubricate when the engine starts to run again. The oil should have a sufficiently low pour test so that it will remain fluid at the lowest temperatures encountered.

The Engineering Research Division of the University of Texas reports that recently it was requested to investigate the quality of the concrete foundation that a central station had put in for a new 750-hp. Diesel engine. It was found that the concrete, which was about three months old, had a compressive strength of 720 lb. per sq.in. The mix was found to have been made of one part of cement to eight parts of pit-run sand and gravel. A test in the laboratory showed that by screening this pit-run material over a ¼-in. mesh screen and recombining the fine and the coarse materials so as to produce a 1:3:5 mix, the strength of the concrete at 28 days was increased 128 per cent over that obtained from a 1:8 pit-run mix, while the quantity of cement per cubic yard of concrete was increased only 20 per cent.

Silences Suction Noises of Air Compressors

An interesting noise problem was recently solved at the water-pumping plant of the Borough of Allenhurst, New Jersey. At this plant compressed air is forced into the top of standpipes to furnish the pressure for the distribution of the water supply of the borough. Two large, electrically driven compressors are in use, and the "snore" of the suction of these compressors was so loud that legal complications were threatened. Two six-inch Maxim silencers were obtained and fitted to the outdoor ends of the suction pipes. The troublesome noise was entirely overcome and with no "wiredrawing" of the intake air.

The air flow in this case was inward from the atmosphere toward the intake-valve chambers, while the noise-wave movement was outward from the intake valves toward the atmosphere. Thus the air was going one way while the noise waves were coming the other way. This condition is, of course, very different from what happens in the exhaust of an oil engine, where these silencers are most frequently used. In the case of the oil engine both the noise waves and the gas flow of the exhaust are outward from the cylinder to the atmosphere.



SHOWS SILENCERS INSTALLED ON THE AIR-INTAKE PIPES OF COMPRESSORS

In the case of an air-compressor suction the noise is caused by the fluttering of the intake valves. The latter are usually spring-seated, and during the suction stroke they lift and seat, or "flutter," which gives rise to a very loud "snore." This "snoring" noise travels out from the place of origin at the velocity of sound, which is approximately 1,100 ft. per sec., and against the current of incoming air. The latter has a velocity of only approximately 200 to 400 ft. per sec., so that the sound waves have no difficulty in gaining the outer air.

In a discussion on Planimeters appearing in the May 2 issue of *Power* the statement was made that the Coffin averaging instrument reads average height only and will not measure areas. This is incorrect as the original Coffin machine and the improved design, known as the Ashcroft, measure both heights and areas.

Indicator Diagrams of Compound Engines

COMPOUND engines include all types of engines where the expansion is carried out in more than one cylinder. However, among engineers the word compound is often limited to those engines where the expansion is divided into two stages. Other stage expansions are triple- and quadruple-expansion engines.

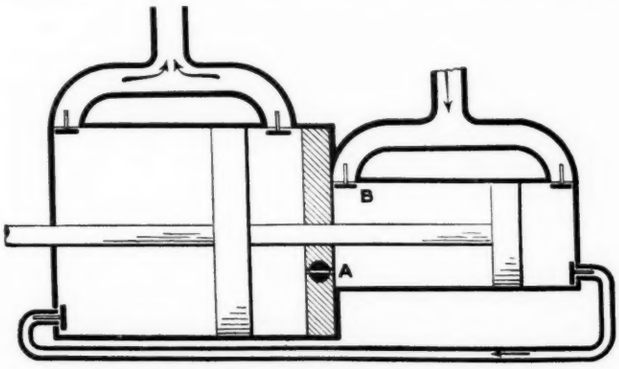


FIG. 1. SCHEMATIC ARRANGEMENT OF TANDEM-COMPOUND WITHOUT RECEIVER

The compound engines are classified as with receivers and without receivers, the latter being the Woolf engines, brought out in Germany and no longer used to any extent. In the United States the best-known engine without receiver was the single-acting compound engine. Certain tandem-compound high-speed engines have very small receivers, but still do have low-pressure admission and exhaust valves.

The compound engines may be separated into the tandem- and the cross-compound groups. The tandem engine, Fig. 1, has the two cylinders mounted on one frame, a common piston rod being used. Obviously, the

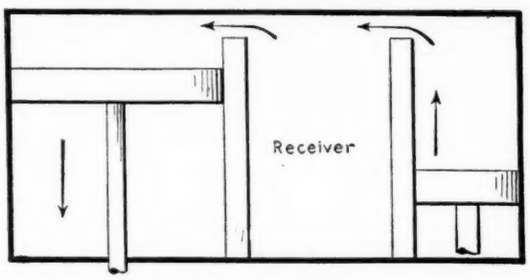


FIG. 2. CROSS-COMPOUND WITH RECEIVER AND CRANKS AT 90 DEGREES

piston position in the respective cylinders are identical for any crank angle.

The cross-compound engine usually has the cranks set at 90 deg., with the high-pressure crank ahead of the low-pressure crank, as in Fig. 2. Other crank angles, such as 45 and 120 deg., are used. The cranks of large engines are never set at 180 or 0 deg., for this causes the maximum pressures to occur in both cylinders at the same time. This would give rise to large variations in the torque exerted on the shaft, and a large flywheel would be required to minimize the effect of these large fluctuations. A number of English high-speed engines do, however, have the cranks set 180 deg. apart, but these engines have the cylinders side by side and the crank arrangement makes a better-balanced engine.

The triple-expansion engine is built as a three-cylinder unit or with one high, one intermediate and two low-pressure cylinders. This latter arrangement decreases the weight of the low-pressure element and by proper arrangement of the four cylinders the pressures in the engines are better distributed.

The type of compound engine has a considerable effect on the shape of the indicator diagram. For this

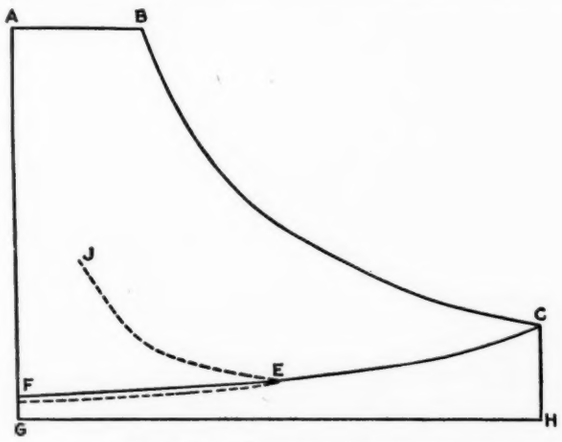


FIG. 3. DIAGRAMS OF TANDEM-COMPOUND WITHOUT CLEARANCES

reason it is well to study the prevailing types and see what manner of diagram is to be expected. In Fig. 1 is shown diagrammatically a tandem-compound without clearance and without a receiver. If steam be admitted into the high-pressure cylinder which is here taken to have a total volume of one cubic foot, through the steam valve *B*, Fig. 1, the piston will be forced to the

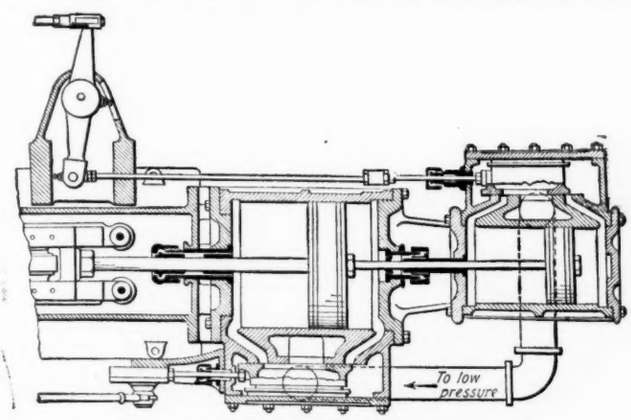


FIG. 4. TANDEM-COMPOUND ENGINE

right giving the steam and expansion lines *AB* and *BC* in Fig. 3. In the figure the high-pressure diagram has been reconstructed to correct for the spring scale. At the end of the stroke we have one cubic foot of steam in the high-pressure cylinder. The pressure at *C* may be found by the $PV = constant$ equation as follows: If

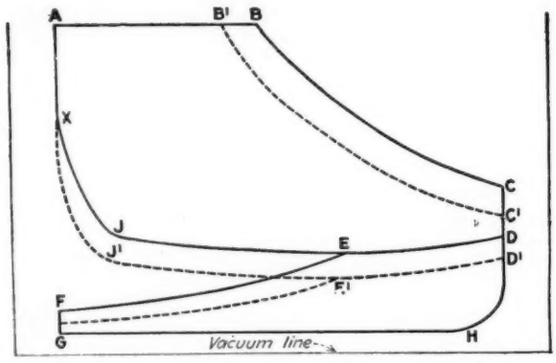


FIG. 5. DIAGRAMS OF TANDEM-COMPOUND

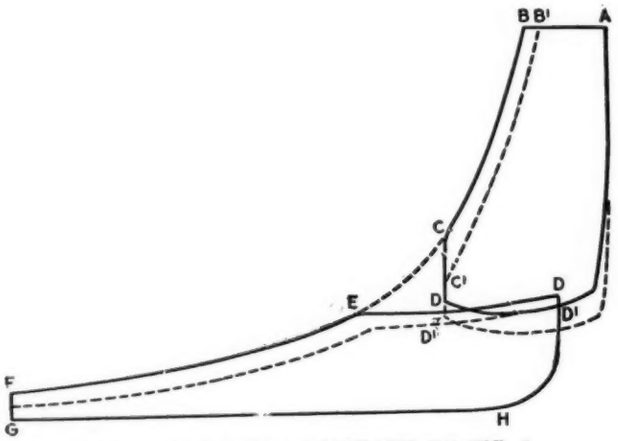


FIG. 6. CORRECTED DIAGRAMS OF FIG. 5

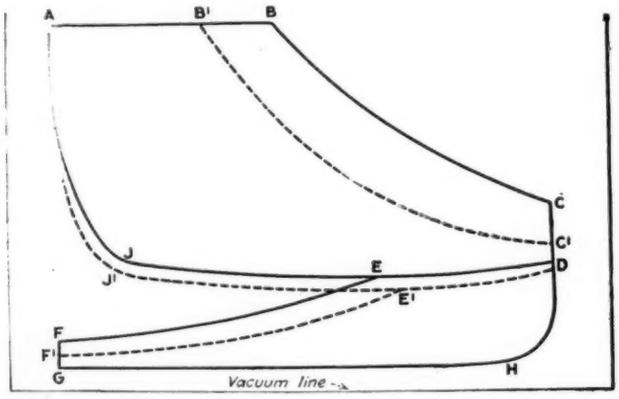


FIG. 7. CHANGE IN LOAD BY VARYING BOTH CUTOFFS

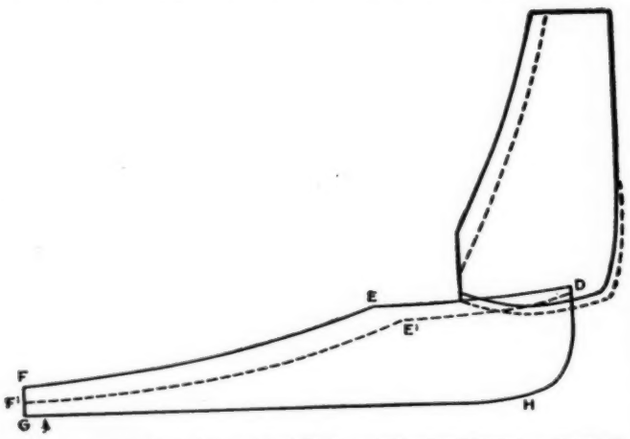


FIG. 8. EFFECT OF VARIABLE LOW-PRESSURE CUTOFF

cutoff occurs at $\frac{1}{4}$ stroke, there is $\frac{1}{4}$ cu.ft. of steam admitted at the initial pressure. Assuming this to be 120 lb. absolute, we have $P_BV_B = P_CV_C$, or $120 \times \frac{1}{4} = 1 \times 30$. The pressure at C is, then, 30 lb. absolute. Assume the valve A to be opened, allowing the steam to enter the low-pressure cylinder. When the pistons have moved one-fourth of the return stroke, we have in the high-pressure cylinder $\frac{3}{4}$ cu.ft. of steam and in the low-pressure a volume equal to one quarter of its stroke volume. If the cylinder ratio is 4 to 1, the total low-pressure cylinder volume is four times that of the high-pressure cylinder, or $4 \times 1 = 4$ cu.ft. Since the piston has moved one-fourth of its stroke, the space in the low-pressure cylinder filled with steam will be $4 \times \frac{1}{4} = 1$ cu.ft. We have then a total volume in both cylinders of $1 + \frac{3}{4} = 1\frac{3}{4}$ cu.ft. The pressure at this point in the stroke, since it was 30 lb. at a volume of 1 cu.ft., will be $30 \times 1 = 1\frac{3}{4}\ P$, or the pressure will be 17.1 lb. absolute. Following the same method, at half-stroke the total volume is $\frac{1}{2} + (4 \times \frac{1}{2}) = 2\frac{1}{2}$ cu.ft. The pressure at this point in the stroke becomes $30 \times 1 = 2.5\ P$, or $P = 12$ lb. absolute. At E, the end of the stroke, we have in the low-pressure cylinder 4 cu.ft. and no space in the high-pressure cylinder. The pressure at this point would be $30 \times 1 = 4\ P$, or $P = 7.5$ lb. The opening of the low-pressure exhaust valve allows the pressure now to fall from E to G and on the return stroke of the low-pressure piston the line GH on the low-pressure diagram is drawn.

Suppose at the half-stroke, or E', the valve A is closed. The pressure at this point, as before, is 12 lb. and the high-pressure piston will continue to compress the half volume of steam trapped in this cylinder, giving a compression line shown dotted in Fig. 3 EJ. The 2 cu.ft. of steam trapped in the low-pressure cylinder will expand and at $\frac{3}{4}$ stroke will fill a volume of $4 \times \frac{3}{4} = 3$ cu.ft. Its pressure will be $3\ P = 12 \times 2$, or $P = 8$. At the end of the stroke the pressure will be $4\ P = 12 \times 2$, or $P = 6$ lb. absolute. This expansion follows the dotted line in Fig. 3 and shows the effect of low-pressure cutoff.

Engines With Small Receivers

Tandem engines as built have a small receiver space, consisting of the connection between the high- and low-pressure cylinders together with the exhaust cavity of the high-pressure cylinder and the steam chest of the low-pressure cylinder. The volume of this receiver space and of the clearances alter the diagram from Fig. 3.

Fig. 4 outlines this construction, and for the purpose of illustrating the pressure-volume conditions it will be assumed that the high-pressure stroke volume is 1 cu.ft., the clearance 10 per cent, or 0.1 cu.ft., the receiver volume 2 cu.ft., the low-pressure stroke volume 4 cu.ft. and its clearance 10 per cent, or 0.4 cu.ft. If the high-pressure real ratio of expansion is 2, then at B, Fig. 5, the volume of steam in the high-pressure cylinder is one half of 1 + 0.1 or 0.6 cu.ft. The initial pressure is assumed to be 120 lb. absolute, and at C, when the high-pressure exhaust opens, this pressure is one half this value, or 60 lb. It will be assumed that the receiver is already filled with steam at 40 lb. absolute pressure. When the high-pressure exhaust opens, we have 2 cu.ft. of receiver steam at 40 lb. and 1.1 cu.ft. of high-pressure exhaust at 60 lb. The two volumes of steam intermingle, with a resulting volume of 3.1 cu.ft. at 47.1 lb. This is obtained by dividing the

sum of the products of the two pressure volumes by the sum of the volumes, as $\frac{(1.1 \times 60) + (2 \times 40)}{1.1 + 2}$ = 47.1.

If the low-pressure admission valve opens at dead center, there is an additional clearance volume that must be filled, lowering the receiver pressure. It will be assumed that this entire clearance of 0.4 cu.ft. must be filled. The pressure in the receiver will drop from 47.1 to the value P (3.1 + 0.4) = 47.1 (3.1) or P = 41.7 lb. absolute. This gives the point D in Fig. 5.

At four-tenths stroke E the admission valve of the low-pressure cylinder closes and traps in this cylinder a volume equal to 0.4 (4) + 0.4 = 2 cu.ft. of steam at 35.5 lb. This steam continues to expand, and at three-

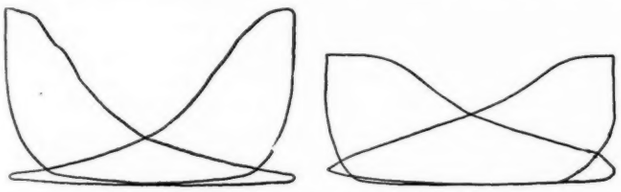


FIG. 9. DIAGRAMS FROM TANDEM-COMPOUND

quarters stroke is at a pressure of (2 × 35.5) ÷ 3.4 = 20.9 lb. When the end of the stroke is reached, the pressure in the low-pressure cylinder has become 16.1 lb. The exhaust valve opens, and the pressure immediately drops to G, Fig. 4, say 8 lb. absolute, and the back-pressure line HG is made on the return stroke.

When the low-pressure admission valve closed, there was contained in the receiver and high-pressure cylinder a volume equal to the receiver + h.p. clearance + 0.6 stroke volume, or 2 + 0.6 + 0.1 = 2.7 cu.ft., and the pressure is 35.5 lb. The continued travel of the high-pressure piston compresses the steam from this volume to 2 + 0.1 + 0.25 = 2.35 cu.ft. at three-quarters stroke. This makes the pressure 41 lb. at this point. The high-pressure exhaust valve in closing at this point traps this steam in the high-pressure cylinder, and as the piston continues to the end of the stroke, this steam is compressed up to the point X where the admission valve opens.

The high-pressure diagram has been corrected for volume in Fig. 6, which shows both diagrams in full lines.

Compound engines are arranged to have a fixed low-pressure cutoff with a variable high-pressure cutoff; many are fitted with governors, which vary the cutoff on both cylinders.

Fixed Low Pressure Cutoff

As example, assume that the engine that gives the diagrams in Fig. 5 has the low-pressure cutoff fixed at ⅘ its stroke. The load drops, and the governor changes the high-pressure cutoff to ⅗ of the stroke. The pressure at the end of the stroke becomes 43.6 lb. The volume at this pressure is 1.1 cu.ft., and mixes with the receiver steam having a volume of 2 cu.ft. and a pressure of 40 lb.; this gives a pressure of 41.2 lb., a trifle lower than in the first case. Upon the opening of the low-pressure admission valve the volume is 2 + 1.1 + 0.4 = 3.5 cu.ft., and the pressure drops to 36.5 lb. The low-pressure cutoff is still at ⅘ stroke, and when the piston reaches the point the pressure has fallen to 27.4 lb. This cutoff in the low-pressure cylinder gives the diagrams the appearance of the dotted outline in Fig. 5. The movement of the high-pressure piston compresses the steam in the receiver and high-pressure cylinder. Since the pressure at E' is less than at E, the receiver pressure at J' will be less than before. On the next stroke the pressure at D' will drop lower than here shown until the correct receiver pressure is established. It will be noticed that the decrease in the high-pressure diagram is practically made up by the lowering of the back-pressure line. If the load is uniformly distributed at a given high-pressure cutoff, then with a shorter high-pressure cutoff the division of the load is changed, the high-pressure cylinder carrying the larger part. On heavy loads the division of the load is such that the low-pressure cylinder carries the larger part. In Fig. 6 the diagrams of Fig. 5 corrected for volume are shown in dotted lines.

Variable Low Pressure Cutoff

If both high- and low-pressure cutoffs are under governor control, another set of conditions exists upon the change of load. Assume that the full-line diagrams in Fig. 7, which is the same as Fig. 5, are taken at a certain load. If the load drops, the governor will make each cutoff earlier. For example, the lighter load may now give a cutoff in the high pressure at B' and make the low-pressure cutoff at E'. The expansion in the

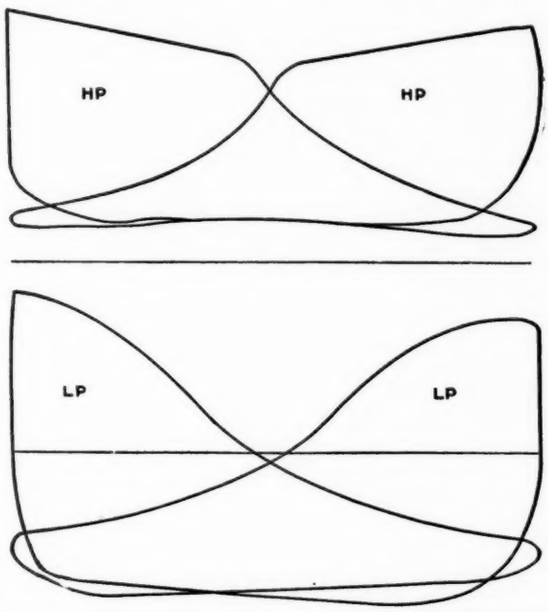


FIG. 10. DIAGRAMS OF CROSS-COMPOUND ENGINE

high-pressure cylinder will carry the pressure to 41 lb. at C', the end of the stroke. The process of computation as outlined before is followed. The pressure in the two cylinders and receiver when low-pressure cutoff takes place is indicated by E'. The movement of the high-pressure piston compresses the receiver and high-pressure cylinder steam to practically the original receiver pressure, as indicated by the dotted line. On the next stroke there is not as low a receiver pressure as in case of the fixed low-pressure cutoff diagram. The corrected diagrams are shown in Fig. 8. It will be observed that with both cutoffs varied both diagrams are decreased in area at about the same rate; that is, the load is still distributed between the two cylinders as before altering the cutoff.

One characteristic possessed by indicator diagrams from many tandem-compound engines is the curved high-pressure back-pressure line, as well as a sloping low-presure admission line. This is noticeable in Fig. 9, diagrams from a 15 x 30 x 18 tandem engine.

With a given cutoff in the high-pressure cylinder, admitting a definite weight of steam, the effect of varying the low-pressure cutoff is to alter the division of load by raising or lowering the receiver pressure. It has no effect upon the number of expansions, which is fixed by the high-pressure cutoff and the cylinder ratio.

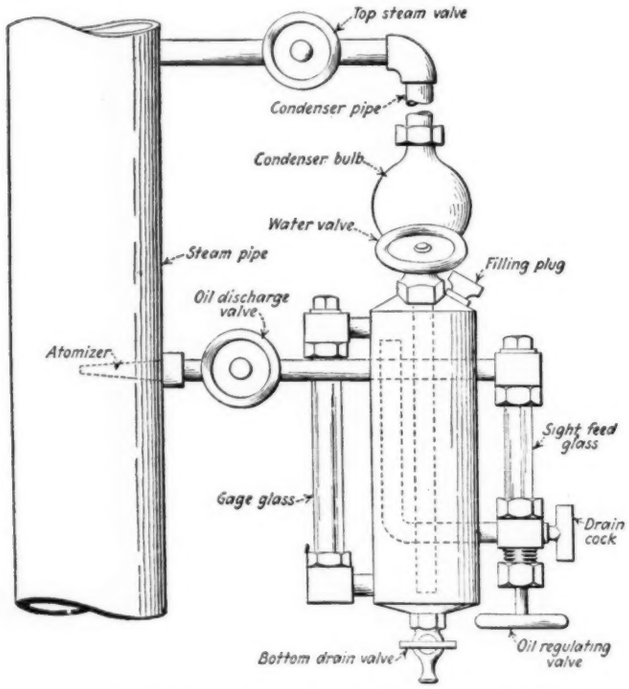
The low-pressure admission valve should be set so that the receiver pressure at the point of high-pressure exhaust-valve closure is the same as the desired receiver pressure at the time the high-pressure steam enters the receiver.

The diagrams from a cross-compound engine, due to the setting of the cranks 90 deg. apart, have the appearance of Fig. 10. Since the high-pressure cylinder is discharging into the receiver before the low-pressure admission valve opens, there is a considerable variation in the receiver pressure during the stroke unless the receiver is large.

Pointers on Lubricator Operation

BY ALLEN F. BREWER

The location of a hydrostatic lubricator with respect to the unit it is to lubricate should be at a convenient height above but close to the throttle valve. The independently acting type of lubricator is attached to the steam line at two points, one above the apparatus itself and one at the point of discharge. The length of the



HYDROSTATIC LUBRICATOR AND CONNECTIONS

pipe connecting the upper attachment to the condenser bulb should be at least eighteen inches, in order to insure sufficient water pressure acting upon the oil within the lubricator reservoir to cause constant and regular flow of the oil through the discharge pipe.

When filling a lubricator, first close the valve in the oil discharge pipe and the valve just below the condenser bulb. Then remove the filling plug and drain the lubricator by the bottom drain valve, before putting in new oil.

To put the lubricator in operation, open the valve in the oil discharge pipe, and also the water valve and then adjust the flow of oil by means of the regulating valve below the sight-feed glass. The regulating valve should not be opened for at least five minutes after steam has been turned on. This will allow sufficient steam to condense in the lubricator to fill completely the reservoir space not occupied by the oil. If this precaution is not taken, steam will enter the lubricator and "churning" will occur. This churning will deposit globules of oil on the sight-feed glass.

If the oil drops appear to form too large, they may be reduced by filling the sight-feed glass with a solution of water to which about a teaspoonful of common table salt has been added. Increasing the specific gravity of the solution in the glass causes the oil drop to leave the feed cone before it has time to become too large. The formation of large oil drops is usually due to dirt collected around the top of the oil-feed cone.

The sight-feed glass can be cleaned of dirt or oil deposits either by blowing steam through it or by swabbing out with a piece of cloth. In the latter case remove the plug at the top, wrap a piece of lintless cloth around a thin stick, soak in kerosene, and swab the sides of the glass. Be careful not to allow any shreds to remain in the glass.

When it is desired to clean by blowing through with steam, close the oil-regulating valve below the sight-feed glass and then open the sight-feed-glass drain cock in the bottom setting. Then by opening the valve in oil-discharge pipe, steam can be blown through the sight-feed glass to clean it.

In general leaky connections will cause irregular feeding of the oil. Should the sight-feed glass become clogged, it may be due to setting up too tightly on the bottom nut when the glass was installed. This often causes the rubber gasket to be squeezed out of place and perhaps into the glass. If the oil-feed cone should become displaced or off center, it may cause the globules of oil to move up the sides of the glass, and perhaps to stick there, causing the glass to fill with oil. The glass should then be removed and the cone filed down, or it will be found effective to solder a fine copper wire to the side of the cone, leading it upward through the center of the glass, nearly to the top. The oil drop will then follow the wire, and the glass will remain clean. It is not advisable to allow a hydrostatic lubricator to run empty of oil, owing to the danger of the sight-feed and gage glasses becoming hot, thus causing oil to stick and gum to the sides of the glass.

When a lubricator is installed where it is exposed to the weather, and there is danger of freezing, it should be shut off and thoroughly drained when not in use. When this is necessary an extra valve should be installed in the upper piping, close to the main steam pipe. All valves should be closed tightly in order to prevent leakage of steam and resultant condensation in the lubricator.

When a small, slow feed is required, it is often found difficult to maintain this uniformly owing to the changes in the viscosity of the oil by virtue of changes in temperature. Therefore, weather or engine-room temperatures may be factors that will necessitate more attention being given to the operation of the lubricator.

EDITORIALS

Watch the Traps

"LOOK out for the pennies and the dollars will take care of themselves."

One of the easiest and surest ways to save pennies and dollars around the average power plant is to make a practical study of traps and apply the knowledge thus gained. A trap seems a small and insignificant piece of apparatus, particularly when it is out of sight in some dark hole, yet a single one, if not in good order, may easily waste several dollars worth of steam per week, while the loss for the whole plant may come to an appalling total.

Again a trap may leak no steam but cause damage to other apparatus because it is partly clogged or is too small to handle the condensation during certain periods. In trying to avoid the installation of too small a trap the engineer may go to the other extreme and waste money by getting one that is much larger than necessary. The scientific thing to do is to determine the maximum amount of water to be handled and the effective pressure available for its discharge. To select a trap on the basis of pipe size alone is rarely justified.

Every engineer should pride himself on his ability to choose traps intelligently and install them properly under any conditions ordinarily found around the plant. Of equal importance is the matter of inspection. This is best handled by strict adherence to a schedule.

All of these matters and many others are covered by E. Smiley in an article in this issue entitled "Steam Traps—Their Selection, Installation and Upkeep." Of particular value is the method of ascertaining whether there is a leakage of steam. In the absence of a gage glass one is often at a loss to know whether the vapor rising from the discharge is partly steam that has leaked through or consists entirely of re-evaporated water. The test described, which can ordinarily be completed in less than half an hour, definitely settles the matter.

Technical Education and Specialization for Engineers

ALTHOUGH many prominent Americans, both engineers and otherwise, are not college or university graduates, there are comparatively few young men of today who spend much time in debating whether a college education is worth while.

Manufacturing, selling, inventing and other branches of business require analysis, science, and logical processes to such an extent that education is usually the first requirement for a position. Industries encourage and co-operate with educational institutions, on which they depend for certain types of employees.

Education for engineers received the consideration of some of the leaders in the profession at the June meeting of the American Institute of Electrical Engineers. Although business inclines more and more to specialization it is noteworthy that highly specialized courses are not recommended. Certain subjects are undoubtedly suitable for electrical engineers, and special post graduate courses are suggested in one particular instance, for a particular field of endeavor. In general, it appeared that broad fundamentals were more important, and that a few subjects be learned thoroughly, rather than a number of subjects superficially. A training which would lead to proficiency in applying mathematics to practical problems, such as analytical geometry and calculus, was stressed.

This is somewhat unexpected advice in regard to the training of a specialist, the name itself implying a confining of activities and a narrowing of outlook. It is further stated that the average college man would be preferred that is normal physically, morally and mentally, without regard to his strength in other qualities. The college should offer the average technical student as suggested by Mr. S. E. Doane, chief engineer of the National Lamp Works, Cleveland, Ohio, training in six directions:

1. Instruction in engineering knowledge. This might be subdivided into the teaching of (a) fundamentals, and (b) specific applications.
2. Instruction in non-engineering subjects, such as English, Economics, Law, etc.
3. Instruction and training in hygiene.
4. Inculcation of habits of clear thinking, concentration, persistence, observation, decision, imagination, etc.
5. Infusion of principles of fairness, unselfishness, tolerance, refinement, courtesy, etc.
6. Formation of friendships.

Another condition for leaving specialization till later was voiced by Mr. Doane, who said: "No student really knows what he will do till after he leaves school. Those who specialize in school may find themselves doing something entirely different within a short time of their graduation." We know also, that if a special industrial subject be taught, it may be hopelessly out of date a few years later, and require more application.

Manufacturers prefer to develop employees in accordance with their own special conditions, so as to function as a part of their organization, and the individual of normal general training forms the best material for molding.

The path of education appears to lie among broad and practical fields rather than technical and special. A man's value to industry, generally speaking, depends on many more qualities than proficiency in one or more scientific subjects. Physical culture, or gymnasium work, is given credit toward a degree by some institutions. Why not take advantage of the much more broadening experience of competition in college athletics and encourage such participation with a recognition of credit? Other activities of college life offer advantages in the development of qualities essential to usefulness and success. Perhaps some day a college degree will mean vital, all around equipment rather than knowledge of specific subjects. Experiments are often undesirable, but the progress of educational institutions, in spite of their physical and financial limitations, will be watched with interest.

Some Form of Development Assured

AS MATTERS now stand, continuation of work on the Wilson Dam is assured, beginning October first and continuing until July first of next year, with $7,500,000 available to the Army engineers for these operations. Thus it would seem that the Government is at last finally and definitely committed to the completion of the project, which promises to be one of the greatest contributors to industrial development of the South.

However, Congress has adjourned without giving any intimation of its intention with respect to the Ford offer, and it is certain that this matter can now receive no attention until Fall. When the House of Representatives reconvenes on August 15, or at a later date if the adjournment period should be extended, the tariff bill will have right of way; then will follow what promises to be a protracted discussion of the soldiers' bonus. That being the case, it is not at all unlikely that Muscle Shoals will fail to receive serious attention until after all the primary elections have passed. If this proves to be the case, the likelihood of Mr. Ford's offer being accepted will be lessened.

It may be that some variation in the Ford proposal will ultimately be accepted. It may be that the Alabama Power Company will be given control of the power generation and disposition under Federal and state regulation. It may be that the Government will decide to operate and sell the power from there as a publicly controlled and operated plant. Whatever may be the final solution, engineers may take comfort in the knowledge that some sort of a Muscle Shoals power development will be had.

Efficiency Versus Economy

IN STEAM power-plant design and operation, among the many factors that must be considered, there are two that must always be kept in mind—economy in heat units and economy in the cost per unit of power produced. In the former the highest plant efficiency will be attained when a minimum number of heat units are expended per unit of power delivered at the busbars. On the other hand, the most economical plant from a monetary point of view is the one that will deliver a kilowatt-hour on the busbars at the minimum cost. One may be considered as economy in the use of our fuel resources and the other as economy in dollars and cents to the power consumer.

Much has been said about efficiency in power-plant operation as a means of conserving our fuel resources, but when it comes to the facts of the case it is a question of dollars and cents and not conservation of fuel. If it were only a matter of fuel conservation, every water power within transmission distance of a load should be developed. But before a water power can be considered economical to develop, it must be able to deliver a unit of power to the consumer at a cost at least as low as a steam plant serving the same load. If the water power cannot compete with the steam plant on a pecuniary basis, its development must wait until the cost of fuel brings about this condition. Of course there may be some exceptions to this rule, where the uncertainty of fuel supply or other factors have eliminated the competition of the steam plant.

The higher the vacuum maintained on a steam turbine the higher its thermal efficiency, but the most economical point as to cost of power is around twenty-nine inches, and this has been adopted as practically standard, although varying slightly with the temperature of the circulating water. Burning pulverized coal under boilers, the use of economizers and preheating the combustion air will give the highest boiler efficiency, but the saving in fuel must be sufficient to pay a reasonable return on the investment before such installations can be justified. In general the highest boiler efficiency will be obtained around normal rating, but the lowest cost of producing steam during normal operation is somewhere around one hundred and eighty to two hundred per cent normal rating and three hundred to four hundred per cent on peak loads even though the efficiency may be considerably reduced at the high ratings. The higher the steam pressures and temperatures, the higher the thermal efficiency of a given plant, but many large companies consider that when all factors are taken into account, the most economical temperature at present is around six hundred degrees, even though plants have been designed for considerably higher temperatures.

Fortunately, the plant that will produce a unit of power at minimum cost is also fairly economical in the use of fuel. However, maximum plant efficiency is obtained at a somewhat higher cost per unit of power than the minimum. Since the public is not inclined to take on a financial burden to conserve our natural resources for future generations, modern power developments must be carried on with economy in mind rather than plant efficiency.

Many a good man, who knew his business, has failed to have the influence his knowledge deserved, for the simple reason that he was unable to get his ideas "across." There are many elements in persuasiveness, but the most important is ability to make the other fellow understand clearly what is in one's own mind. Among engineers, at least, this probably counts for more than any subtle "influence" that a man may have over his fellows. Most of us are "eye minded." We understand a thing best when we see it. The next best thing is a good sketch. For this reason every engineer will find it worth while to get the knack of making simple free-hand sketches of apparatus, tools, etc. These need not be artistic nor even neat. But they should serve the valuable purpose of showing the other man just what is in the mind of the man that draws them.

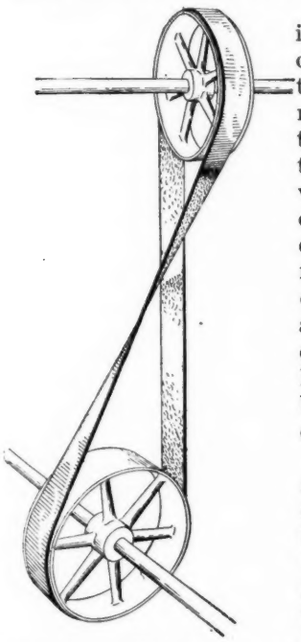
Much has been said about advertising the engineer to the public, but much remains to be done. Stationary engineers, in particular, could certainly benefit by a more general public appreciation of the fundamental importance of their work to industry and civilization. The trouble is that the average operating engineer is a modest man, one who dislikes to parade his performances before the eyes of others. Association with his fellow engineers in such an organization as the N. A. S. E. should help him to overcome some of his natural reticence to the end that business men and public officials may better appreciate the value of the service he renders.

In plant maintenance, as in everything else, no perfect system has been devised. Different conditions require different treatment. Changes in equipment or plant expansion may create conditions that render previous methods of handling maintenance work inadequate and require reorganizing to meet new demands.

PRACTICAL IDEAS from PRACTICAL MEN

Quarter-Twist Drive That Increases Life of Belt

The cut here shown illustrates a quarter-twist drive, with a half turn in the belt, on one side only.



DRIVING SIDE OF BELT REVERSES EVERY REVOLUTION

By referring to the cut it will be seen that the side of the belt in contact with the pulley reverses every revolution of the belt. This tends to increase the life of the belt materially and prevents it getting longer on one side than the other. On drives where two belts are run close together, the tendency for the belts to climb and wear the edges is eliminated. The wear on the lacing also takes place on both sides, therefore it lasts considerably longer.

I have had many years' experience with drives of this kind and have used both leather and rubber belts with good results. The pulleys should be aligned on the shaft the same as for any quarter-twist drive.

O. H. BOLTZ.

Charlottesville, Va.

How Fire in Coal Storage Was Stopped

The public utility by which I am employed makes a practice of keeping 90 days' supply of coal on hand. The fuel used is 1¼-in. screenings, and about 100 tons is required for an average day's run.

For a number of years we experienced trouble and considerable loss from spontaneous combustion. It was necessary to keep rehandling the coal to keep the fire under control. The coal was received in cars and shoveled into wheelbarrows and wheeled out over the pile on an overhead run. It was reclaimed by shoveling into a wagon and drawn to the boiler room. This proved an expensive method of handling.

As the plant was equipped with a revolving crane for handling coal directly from the cars to the overhead bunkers, it was evident we could make a big saving if we could store our reserve pile so that it would not catch fire. It was noticed that the fires always broke out on the side of the pile where the lumps rolled down on the outside and separated from the fine coal. From observations made it was readily seen that by placing the coal on the pile in such a way that it would not separate, we could overcome the trouble.

For the last two years we have handled our storage pile in the following manner: The coal is received in dump-bottom cars and dumped in shallow pits beside the track, then drawn back onto the storage pile with a team of horses and a drag scraper. This method, although more expensive than putting in storage with a locomotive crane, has two advantages: First, the coal lies wherever the scraper is dumped and has no tendency to run and separate; secondly, the constant tramping of the horses packs it together so tightly that there is no chance for the infiltration of air.

We had a pile of 9,000 tons of coal stored in this way over a year, and we have burned more than one-half of it and found no fire as yet.

It costs about 25 cents a ton to store and a similar amount to reclaim, but since it is possible to keep a supply in storage indefinitely this is not to be compared with the loss from fire when stored with the traveling crane.

E. K. MANNING.

West LaFayette, Ind.

Filing and Indexing Clippings

The methods of filing clippings seem to be as numerous as the persons who have use for them, so here goes for another, which, while one of the oldest, seems to be less used for technical clippings than any.

I have used the envelope method for taking care of clippings but never was satisfied with it. The large size of a number of the technical periodicals requires an equally large envelope unless the clipping is folded, and it does not take long for a folded paper to become two or more papers. Frequently, an article will fill one or more pages and end with an inch or two on another page, and it is not easy to insure the attachment of the fragment to the whole so that it will stay.

Then there are articles only an inch or two in length that are decidedly worth while preserving, but they have a habit of crawling to the bottom of the envelope or getting sandwiched in a folded article.

In my opinion a scrapbook has many advantages over the envelope system. I use two sizes with numbered pages, one of them large enough to take a page of the larger periodicals, in which I paste the clippings (not always at the time that I clip them, but as opportunity offers). Where the article is all on one page, I paste at the two side edges; if it is on more than one page, I paste at the inside edge only. If the article ends on a portion of a page, I paste the portion on the scrapbook leaf, under the last page of the article.

If a clipped leaf has two articles on it, one on each side, the one that is uppermost when pasted in the book (by the inside edge) is indexed by the page number, and the one underneath is indexed by the same page number, with the fraction ½ added. For example, 98 for the upper and 98½ for the under article.

The other size scrapbook is a type of self binder, the kind that has two cords fastened to one cover which are passed through holes on one edge of the sheet to be bound and then through holes in a metal strip fastened to the other cover. This binder is of a size to take the standard 9 x 12 page, and no other size is bound in it, so there is no pasting at all. I simply take out certain pages of a periodical, punch two holes through the inside edges, thread the cords through them and tie them up. If an article does not use a whole page, the whole page is bound, and pen or pencil lines drawn through the rest of the paper. The pages are numbered serially in the book, disregarding the printed numbers of the periodical pages. The books are numbered as volumes, and the card index refers to an article as in Scrap Book, Volume ...., Page .....

Philadelphia, Pa. A. G. COLLINS.

Engine Operated Totally Submerged

There has been considerable discussion in *Power* from time to time on the wrecking of engines by water.

An instance where a high-speed compound engine continued to run for 2½ hours submerged under 35 ft. of water, was described some time ago in the *Times Engineering Supplement*, published in London. According to the report, in a Canadian city the pumping station and power plant are situated in a river valley several feet above the normal flood level. The steam for the pumping plant was supplied from the power plant about 500 ft. away. The steam pipe was laid at ground level and housed in a wooden casing of dry sand. It supplied steam to a large high-speed vertical inclosed engine driving a high-lift centrifugal pump and two smaller engines of similar type coupled to low-lift centrifugal pumps, the latter being placed in a pit about 40 ft. deep.

To the steam main supplying these engines was connected a steam-flow meter, a chart from which is shown in the illustration.

Some years ago, during the flood period, the river rose to a height unprecedented in the history of the city, and it soon became evident that the water supply and possibly the light and power service would have to be suspended.

Reference to the illustration will show that the meter recorded a sudden demand of the pumping plant for steam during the afternoon, the flow increasing from about 14,000 lb. per hour at 1:45 p. m. to nearly 20,000 lb. per hour at 2 p. m. All this extra steam was taken by the high-lift pumping engine in response to the call for more power to supply the sudden drought on the water-mains caused by the filling of all available vessels in readiness for a cessation of the supply. By this time the water was spreading over the flats and had reached the casing of the steam line to the pumphouse. It rapidly found its way to the surface of the pipe, and the condensation caused is reflected in the rapid rise in the quantity of steam that had to be supplied.

Between 4 and 5 o'clock the water began rapidly to fill the low-pressure pump pit and to submerge a Belliss compound engine of about 150 hp., driving a centrifugal pump which was raising the low-lift water at the time. As there were two other low-lift pumping sets, giving ample reserve capacity, it was considered advisable to leave the Belliss engine running as long as it would continue to do so, because it was of the highest importance to keep the water in the intake pipes in motion while so much mud and silt were in suspension. Hence the engine was deliberately offered as a sacrifice to the Gods of the River, and it actually continued to run, totally submerged under many feet of muddy water for something like two hours and a half. The drop in the steam demand as shown on the diagram indicates that it broke down at a few minutes before 7 p. m., this time agreeing with that at which its atmospheric exhaust ceased. The cause of the stoppage proved to be the breaking of the connecting-rod bolts, the engine otherwise being little the worse for its ordeal.

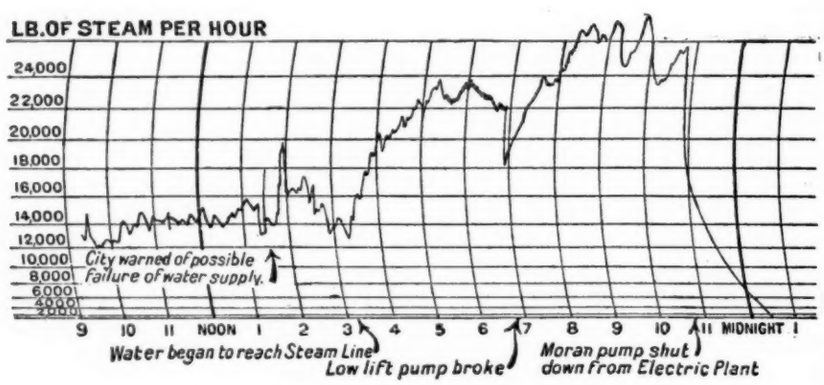


CHART FROM FLOW METER SHOWING INCREASE IN STEAM CONSUMPTION

A further submergence of the steam pipe by the still rising waters is reflected by the rapid and continuous rise in the steam consumption, which had exceeded 26,000 lb. per hour on the limits of the diagram by about 8:20 p. m. More than half of this steam was lost by condensation in the submerged pipe, as the only engine on the line at that time was the larger Bellis set driving the high-lift pump. This set was "alone in a wide, wide sea," the engineer standing on the crank casing to keep himself out of the water. He was rescued by a boat, and the engine that was left running to maintain the city water supply was eventually shut down just before 11 p.m. by turning off steam from the line from the main boiler house. Both boiler-room and engine-room basements in the electric plant had, of course, long been flooded, but word was not given to draw the fires until the water had crept up to within a few inches of the mud-drums of the boilers.

Birmingham, England. A. C. PAIN.

Putting Steam Turbines in Service

The article in the May 16 issue, entitled "Putting Steam Turbines in Service," is both interesting and instructive. It describes in part just what will happen when a certain operation is made whether it be right or wrong. The manufacturer does not always go into detail in his instruction sheet as to why you should or should not do a certain thing and what might result from the neglect of carrying out his instructions to the letter.

After installation of a turbine, it is not unusual for the operator to read the instruction sheet over a few times, acquaint himself with the different valves and their operation and then hang the valuable instructions in a remote place on the wall or place them in an unfrequented drawer of his desk, to be consulted only in time of trouble, which is generally too late to avoid disastrous damages.

Temperature changes are some of the most important problems the builder has to cope with in the manufacture of small as well as large turbines. The average engineer has no conception as to how rapidly these changes take place on a large machine, or he would be more cautions in its operation. It is interesting to note the fluctuations of the thermometers placed on the low-pressure end of the turbine cylinder when a large unit is being put through the third degree by the manufacturer. The cylinder in some places is about 4 in. thick—still a thermometer will actually hop up and down considerably just after the machine is brought to speed and load is being applied at the rate of about 5,000 kw. every ten minutes. The so-called "third degree" is a process of temperature changes the manufacturer puts the machine through after erection. The machine is run at a low speed, and the vacuum is varied periodically from zero to a high vacuum and vice-versa, all running conditions and temperatures being noted during these periods. A high vacuum is suddenly broken and as suddenly applied to the machine to determine whether the distortion of the stationary and moving elements which takes place with these changes will cause rubbing in any extreme. It will be appreciated by your readers what might happen if the vacuum—say 28 in.—and all temperature conditions corresponding to a 28-in. vacuum throughout should suddenly drop to zero with a load on the machine. This sudden change in temperatures would have a tendency to distort the turbine in almost every direction. With the necessarily close running clearances on the modern turbine, this distortion would in all probability cause severe rubbing and consequent damage.

The "rough spots" in the record of large turbines have, no doubt, taught us all that the utmost care must be exercised in the handling of these large units. Central stations have found from costly experiences that these large machines cannot be used for stand-by service, to be started and put on the load at short notice with not sufficient time for the metal to adjust itself to a safe operating condition.

Allowing one minute per 1,000 kw. capacity for starting up and getting the unit into service is not sufficient time for the revolving and stationary elements to expand and go where they will. One minute and a half per 1,000 kw. is a much safer period; applying the load at the rate of 5,000 kw. every five minutes will bring the temperatures back to normal not too rapidly. Almost every turbine has a "critical" period, or period of vibration, while coming up to speed. Experience will teach the operator to avoid running the machine at this speed for too long a time. Some stations make a practice of starting up with from 15-in. to 20-in. vacuum on steam-sealed machines and maintaining that vacuum to three-quarters speed; then allowing vacuum and speed to come up together until the governor takes hold. It will generally be found that after three-quarters speed, if the vacuum is built up, no more steam will be required to bring the turbine to the operating speed.

Too much importance cannot be placed on the too slow opening and leaky throttle valves. Wheels have been known to loosen up on the shaft by escaping steam from a leaking throttle valve. Some central stations have eliminated the danger and anxiety of the starting period on large turbines to a minimum by operating their large units continuously, consistent with good operation, and using smaller units for emergency service.

J. A. DROGUE.

Arlington, N. J.

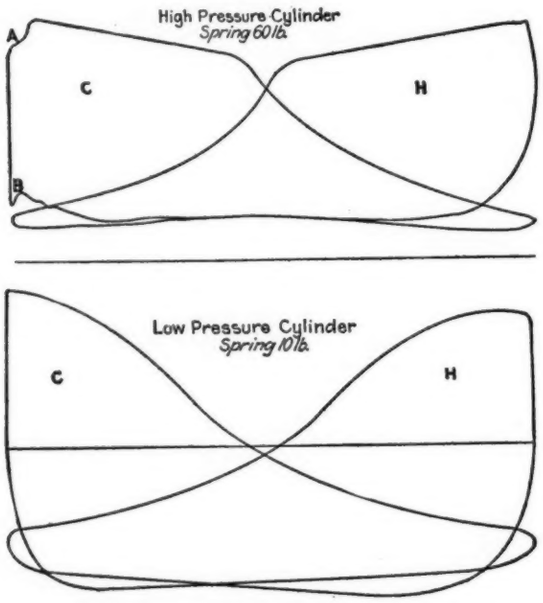
Indicator Diagrams for Criticism

With reference to the diagrams from John R. Spugg in the April 18 issue, I found two 20 x 38 x 48-in. Corliss engines that made very similar diagrams. The hook at the end of compression dropped down almost to the receiver pressure and at both ends of the cylinder.

The engines were used as emergency units in a hydro-electric system and were operated only occasionally. Steam was carried on the boilers at low pressure when the plant was shut down. The throttle valve on the engines leaked slightly and the condensation dropped through the steam ports at both ends of the cylinder. The continuous spattering of the water corroded and made deep cavities in the cylinder bore near both ends. The cavities thus formed caused a bad leak past the pis-

ton when the engine was operated. This may be the trouble with Mr. Spugg's engine.

In the high-pressure diagram *C* the maximum pressure is not reached at the beginning of the stroke due to the leak past the piston. The port opening is also restricted, otherwise the steam line would be nearer horizontal and cutoff would occur earlier, giving a better expansion line. The exhaust line is fairly good, and



DIAGRAMS FROM COMPOUND ENGINE

when the valve closes, compression takes place until the piston comes over the cavity again, at which point the pressure drops as shown at *B*.

The exhaust valves close a trifle too early for good Corliss-engine practice as only sufficient compression is needed to cushion the reciprocating motion of the piston, etc.

The slope in the exhaust line of the low-pressure diagram is caused by restricted opening of exhaust port, and the minimum pressure is not attained until just before the exhaust valve closes. C. F. CLARK.

Buffalo, N. Y.

Comparative Test on Coal and Oil-Burning Boilers

In the June 13 issue, S. A. Smith, Jr., gives a comparative test on coal- and oil-burning boilers.

Taking the tests as they stand, it would appear that oil showed a slight economy over coal, but I believe there should be something added to these figures before we accept them.

It is quite evident that originally there were two boilers and one was changed from coal to oil burning. It must have cost a considerable amount of money to make the change. Should not the interest on this investment as well as the maintenance, taxes, depreciation, etc., be included as a charge against that oil-burning apparatus? What changes were made in the setting to introduce the oil-burning equipment? Is it not true that when the oil-burning equipment was installed, the setting was so modified as to make it an efficient seting for burning oil? Why should we not do so also for coal?

This question is asked because it is evident from the report that as a coal-burning arrangement the setting was very poor, so that there was no reason to expect even with good coal that excellent results would be obtained. It is, however, evident from the analysis of the coal that an evaporation better than 7.96 should be reached, and if it was not, the setting should be modified so that the results to be expected would be maintained—of from 9½ to 10 lb. With this result reached, the figures as to steam cost would be reversed. In my opinion these tests do not give a fair comparison.

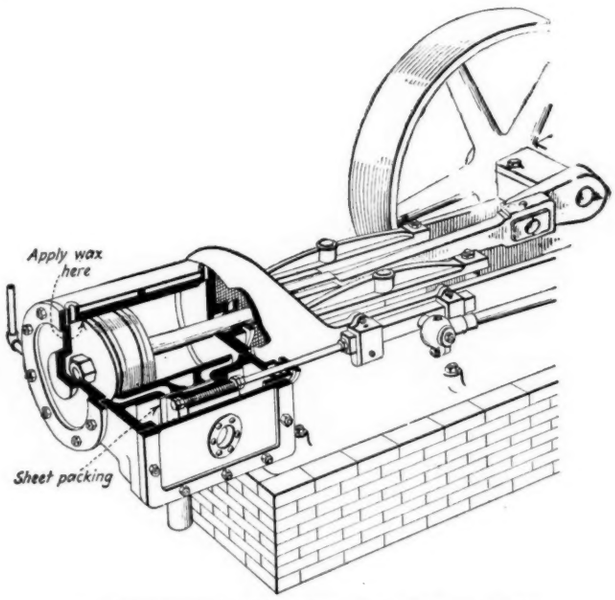
Boston, Mass. HENRY D. JACKSON.

Finding the Clearance Volume of An Engine

In the May 30 issue several methods are given for finding the clearance volume of an engine by filling the clearance space with water.

There are not many engines with piston rings and valves perfectly watertight, and for that reason the clearance volume cannot be determined with any degree of accuracy unless some precaution is taken to make these watertight.

At some time when the cylinder head is removed for inspection, place the piston on the head-end center, then fill the space between the piston and cylinder wall with soft wax, paraffin or soap, also around the edge of the exhaust valves if it is a Corliss engine. This simple method will prevent any water from leaking by. The



PREVENTING LEAKAGE OF WATER PAST PISTON AND VALVE

cylinder head can then be permanently replaced and the clearance space can be filled with water by the usual method.

In the case of a slide-valve engine the valve can be raised slightly and a piece of sheet packing inserted across the ports, as shown in the illustration. The valve can then be blocked and a tight joint insured.

The clearance volume of the crank end will be practically the same as the head end minus the space filled by the piston rod, provided the striking distances are the same.

New Bedford, Mass. R. G. SPOONER.

Economy of Coke as Steam Fuel

In answer to a question in a recent issue of *Power* with regard to the evaporative value of coke, it was stated that it usually contains about 60 per cent as many heat units per pound as the coal from which it is produced. It would be more fair to this fuel to say that the residue from one pound of coal, after the gases have been driven off in the coking process, contains 60 per cent of the heat units that were in the coal originally. If we take one pound of coke, however, and compare it with a pound of coal the case is very different. We should find that, although the relative proportions of fixed carbon and volatile matter have been greatly changed, the B.t.u. amount to about 90 per cent of those contained in a pound of bituminous coal. In fact, the B.t.u. per pound value of good run-of-oven or unscreened gas coke often equals or exceeds that of buckwheat coal.

Good gas coke made by the vertical process shows, upon analysis, 84 to 87 per cent fixed carbon, 10 to 13 per cent ash, around 1 per cent volatile matter and about 0.4 per cent moisture when fresh from the ovens. The heat value is around 12,000 to 12,750 B.t.u. per pound. Generally speaking, bituminous coals have about 5 per cent of hydrogen forming part of the volatile matter. A part of this hydrogen combines with oxygen in the air supplied and forms water vapor and constitutes an inherent loss of about 4½ per cent, or 575 B.t.u. per pound of coal with a flue-gas temperature of 550 deg. F. At a boiler-operating efficiency of 62½ per cent this represents a loss of $\frac{0.625 \times 575}{970.4} = 0.37$ lb. of water evaporated per pound of coal. Partly on this account the ratio of evaporation per pound of coke to evaporation per pound of soft coal is higher than generally supposed, as this loss is practically non-existent with coke.

In burning coke, it is found to be satisfactory in practice to carry a fire from 12 in. to 14 in. thick for the following reasons: (a) To allow for the greater volume of coke containing the same number of heat units as a given quantity of coal; (b) on account of the porous nature of coke a thinner fire would allow too great a percentage of excess air to pass through the fuel bed; (c) a thick fire prevents a too intense rate of combustion taking place immediately over the grate bars and avoids all risk of damage to them.

With natural draft it is best to regulate as far as possible with the dampers, which should be capable of easy and close adjustment, so that the ashpit doors can always be partly open. In burning coal, it is the layer of ash which protects the grate bars, and any coal without ash would soon destroy them. With coke, in many instances, the fuel is more completely consumed than with coal, especially if the latter has a high percentage of ash. With coal the volatile matter, amounting to about 33 per cent in soft coals, is not all burnt, and a portion of it goes up the stack invisible to the eye and also in the form of smoke. On the other hand, the fixed carbon in coke is a substance that can be consumed more completely, and if it is not efficiently consumed the resulting increased proportion of ash is evidence of that fact.

Owing to the small quantity of sulphur in coke there are no troublesome clinkers to be extracted when fires are cleaned. With natural draft run-of-oven coke can be burned at a rate up to 22 lb. per sq.ft. of grate with a stack draft of 0.6 in. of water, equivalent to 150 to 170 lb. of water evaporated per square foot of grate area. Under ordinary forced-draft conditions 30 to 35 lb. of coke can be burned per square foot of grate area, giving an evaporation of 220 to 240 lb. of water per square foot of grate area. Higher rates of consumption per square foot may be attained with increased air pressure.

In buying coke a consumer is purchasing heat under another form, in which the heat units are contained in a much greater proportion of fixed carbon than is the case with bituminous coal high in volatile matter. In so far as the chemical constituents and their relative proportion are concerned, coke belongs to the same class as the anthracite coals, the difference consisting only in physical form. This form has the disadvantage of increased bulk, but has the advantage of enabling the fuel to be burned efficiently with natural draft, the necessary furnace draft being about a quarter to a third of that required for buckwheat. In boiler plants using steam jets to produce draft for burning the smaller sizes of anthracite, this wasteful method of using steam may often be cut out by burning coke, by this means alone effecting a saving of about 8 to 10 per cent.

The extra work in the boiler room is, after all, more apparent than real, and when the firemen are provided with coke shovels of large capacity, the number of shovelfuls per hour is little in excess of that required for soft coal.

One cubic foot of gas coke contains the equivalent in heat units of 340.5 lb. of water evaporated from and at 212 deg. F., and, when burned at an efficiency of 65 per cent, should give an evaporation of 224 lb. of water from and at 212 deg. F., and pro rata, as the case may be. In considering storage space, 16½ cu.ft. per 1,000 sq.ft. of boiler-heating surface should be allowed for. To compare the relative fuel costs of coal and coke, the equivalent evaporation from and at 212 deg. per pound of fuel should first be determined from tests. Suppose this evaporation rate for coal is 9 lb. and for coke 8 lb., and the delivered costs per ton are $7 and $6 respectively. The price of coke should then be multiplied by 9/8, and the saving over coal per ton would be $7—$\frac{\$6 \times 9}{8} = 25$c. The costs per 1,000 lb. of steam can be found simply by dividing the price of each fuel per ton of 2,000 lb. by twice the rate of evaporation attained in each case. Thus, in the case of coke referred to, the cost per 1,000 lb. of steam from and at 212 deg. would be $\frac{\$6}{2 \times 8} = 37\frac{1}{2}$c.

As a check on the figures, the difference in costs per 1,000 lb. of steam multiplied by twice the evaporation rate for coal should equal the saving per ton of coal, or a saving of 25c. effected by burning coke. Gas coke can generally be burned satisfactorily on most grates designed for soft coals. With natural draft the best widths of air spaces are from ⅜ to ⅝ in., depending upon the characteristics of the coke used and the height of stack. Coke breeze requires forced draft and is most efficiently burned on chain grates designed for the purpose. An evaporation of 7 to 8½ lb. of water per pound run-of-oven gas coke may be considered a fair output to be attained in practice with the same degree of care as is generally met with in good hand-fired boiler-room operation.

C. O. THOMAS.

Montreal, Canada.

QUESTIONS AND ANSWERS

Detection of Broken Staybolt

How is it determined whether screwed and riveted staybolts of a locomotive type of boiler have become broken? H. E. C.

Breakage of a hollow staybolt or of staybolts that have their ends drilled with a hole that extends to the fracture is made known by leakage of water. Leakage around the end of a staybolt, when increased by hammering, is an indication of probable breakage, but certainty of fracture can be determined only by drilling a hole throughout the length of the staybolt.

Formula for Length of Open Belt

What is the formula for determining the length of an open belt when diameter of pulleys and distance between centers are given? T. B. N.

The formula for the approximate length of an open belt is

$$L = 2C + 3.1416\,\frac{(D+d)}{2} + \frac{(D-d)^2}{4C}$$

where L = Length of open belt in inches;
C = Distance, center to center of pulleys in inches;
D = Diameter of larger pulley in inches;
d = Diameter of smaller pulley in inches.

Clearance over Bridge Wall of H.R.T. Boiler

In a horizontal return-tubular boiler setting, is 0.3 of the boiler diameter sufficient clearance between the top of the bridge wall and the bottom of the boiler? Where bituminous coal is used, has the flat-top bridge wall any decided advantage over the older type which curved like an inverted arch following the shape of the boiler? W. J. L.

In an ordinary setting of a return-tubular boiler, the furnace bridge wall is for the purpose of preventing the coal from falling off the grates, and for baffling the draft to force the air to pass up through the whole fuel bed.

The shape of the wall, whether flat on top or curved to correspond with the shell of the boiler, or whether the wall is provided with vertical or with sloping sides, appears to make little difference. The area over the wall must be at least large enough not to check the draft afforded by the total tube area. With a flat top for the bridge wall, sufficient area is obtainable when the distance from the top of the wall to the under side of the boiler is 25 per cent of the boiler diameter. A flat wall is easier to build and is preferable to an inverted arch form, as the latter is more likely to cause greater concentration and impingement of the heated gases against a girth seam of the boiler.

Working Pressures for Valves and Fittings

How are valves and pipe fittings classified with respect to working pressures for which they may be suitable? R. L. G.

Valves and fittings are classified, as a rule, under five general headings: Low-pressure, standard, medium-pressure, extra-heavy and hydraulic, which are understood to represent the following working pressures:

Low-pressure—Suitable for working steam pressures up to 25 lb. per square inch.

Standard—Suitable for working steam pressures up to 125 lb. per square inch.

Medium-pressure—Suitable for working steam pressures from 125 to 175 lb. per square inch.

Extra-heavy—Suitable for working steam pressures from 175 to 250 lb. per square inch.

Hydraulic—Suitable for high pressures of water up to 800 lb. per square inch.

Blocking of Governor for Starting up Double-Eccentric Corliss Engine

In starting up a double-eccentric Corliss engine, on which the valves are set to obtain cutoff after one-half stroke, why is it necessary to block up the governor to a position that will cause the cutoff to operate? R. G.

With a Corliss releasing-valve gear the cutoff that is controlled by the position of the governor must occur between the time when the valve opens and when the eccentric has carried the wristplate to the extreme point of its travel to one side or other side of the central position. If cutoff has not occurred within that period, the valve, being unreleased, will not be closed again until the wristplate has returned to the position that it had when the valve began to open. As the period of travel from the central position to the extreme position must occur during one-fourth of a revolution, or about one-half stroke, to obtain cutoff later than one-half stroke the valve must be opened before the eccentric has reached the middle of its travel. In a double-eccentric Corliss engine this is accomplished by setting the eccentric back of the 90-degree position—that is, giving it negative angle of advance—and shortening the steam rods so the valves will have negative lap, or be open when the wristplate is in central position. Hence, if the valve failed to be tripped, it could not be positively closed until after the eccentric had accomplished more than 180 deg. of revolution and the piston had gone a considerable distance on its return stroke. To prevent this occurrence in starting up, the governor must be blocked up in such a position that the cutoff will operate.

Copper Tubes Unsuitable for Water-Tube Boilers

In water-tube boilers would not copper tubes be preferable to those made of iron or steel? E. H. S.

Copper is not suitable because the material cannot withstand the high furnace temperatures common with these types of boilers. Copper is strongest when cold and loses its tensile strength with increase of temperature. At 529 deg. F., which is just above the temperature at which boiler steel and wrought iron attain maximum strength, copper has lost 25 per cent of its strength at ordinary atmospheric temperatures and that is only about half the strength of mild steel in a cold state. The loss of strength of copper diminishes with rise of temperature until at red heat, or about 1,300 deg. F., the strength is negligible. Copper has been used successfully for the smoke tubes of locomotive boilers, but is unsuitable for the tubes of water-tube boilers exposed to forced furnace temperatures.

Arrangement of Adjustable Riding Cutoff

How is a riding cutoff valve arranged so adjustment of fraction of cutoff may be made while the engine is running? W. T.

The lead, exhaust and compression are completely controlled by a main valve which, as indicated in the diagram Fig. 1, consists of an ordinary D slide valve that is extended in length with two admission ports *P* formed through it, each equal in width to the ports in the cylinder face. The main valve is driven by an eccentric, in the ordinary way, and the back of the valve is formed into a valve face on which slides a pair of expansion plates. These plates constitute the cutoff valve. The expansion plates, as shown in Fig. 2, are connected by a right- and left-hand screw which receives its motion from a second eccentric. The function of the plates is to cut off the steam by closing the ports *P*, Fig. 1, in the main valve.

A sectional view of an actual arrangement is shown in Fig. 2, in which *V* is the main valve, connected by its rod *L* to a simple-eccentric valve gear, and V_1, V_2 are the two separate parts of the cutoff valve connected by the right-and-left-hand screw *K*. This screw can be turned from the outside of the steam chest by the handwheel *W*, and is connected to the valve rod of a simple

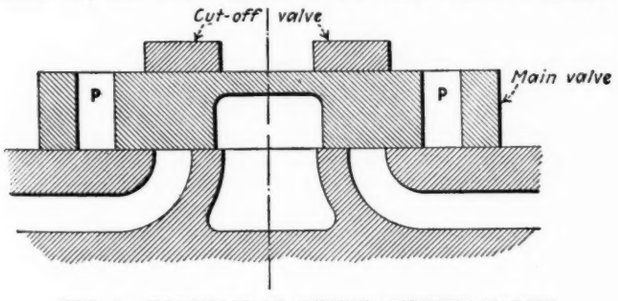


FIG. 1. DIAGRAM OF RIDING CUTOFF VALVE

eccentric gear by a joint that allows the necessary freedom of rotation. The wheel *W* is carried in a sleeve held in a fixed bearing, and the end of *K*, squared, passes through the sleeve. Hence *K* can be turned, and at the same time it is free to move lengthwise through the sleeve. The projecting head of the sleeve is screwed and carries an index nut *I* which serves to indicate how far V_1 and V_2 are separated, and a graduated scale, or index plate, usually is associated with *I* to indicate the fraction of cutoff that is obtained. The cutoff edges of the main valve are at *s* and *S* respectively, and the cutoff edges of the cutoff valve are at *t* and *T*. In Fig. 2 the cutoff valve is shown in a central position relative to the main valve with the main valve ports open by the amount *l*.

If the two eccentrics are of the same throw and same angle of advance, the cutoff valve will ride upon the main valve in the same position. The result then would be the same as though the cutoff valve were a fixed part of the main valve. However, the eccentric for

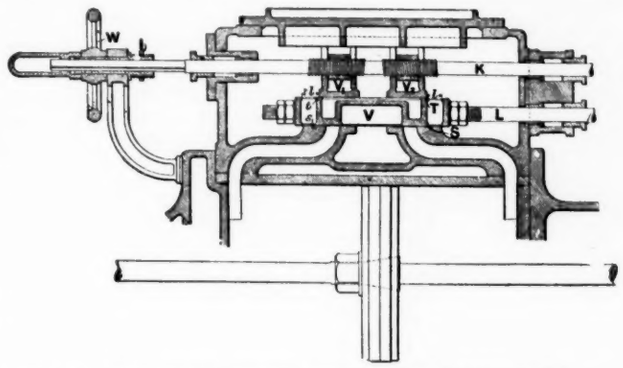


FIG. 2. SECTIONAL VIEW SHOWING ARRANGEMENT OF ADJUSTABLE RIDING CUTOFF

operating the cutoff valve is fixed in a position about 90 deg. ahead of the main eccentric; that is, about 180 deg. ahead of the crank. Hence, when the piston starts on its stroke and the main eccentric starts to open the main valve, the cutoff eccentric moves in a contrary direction and V_1 or V_2 of the cutoff valve gradually closes whichever of the main valve ports *P*, Fig. 1, may be in position for admitting steam from the steam chest to the cylinder. Therefore the farther apart the plates of the cutoff valves are separated by rotating the handwheel, the earlier the cutoff.

Pipe Threads for Boiler Connections

What is the rule for the number of screw threads there should be for a tapped pipe connection to a boiler? R. L.

Threaded openings in a boiler for pipe connections 1 in. in diameter or over should have not less than the number of perfect threads required by the A. S. M. E. Boiler Code, as per the following table:

MINIMUM NUMBER OF PIPE THREADS FOR CONNECTIONS TO BOILERS

Size of pipe connection, in.....	1 and 1¼	1½ and 2	2½ to 4 inclusive	4½ to 6 inclusive	7 and 8	9 and 10	12
Number of threads per in...	11½	11½	8	8	8	8	8
Minimum number of threads required in opening.....	4	5	7	8	10	12	13
Minimum thickness of material required to give above number of threads, in....	0.348	0.435	0.875	1	1.25	1.5	1.625

When the boiler material is not of sufficient thickness to obtain the designated number of standard pipe threads, there should be a pressed steel flange, bronze composition flange, steel-cast flange, or steel plate, constructed and riveted to the boiler so as to give the required number of perfect threads.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Rational Basis for Coal Purchase Specification*

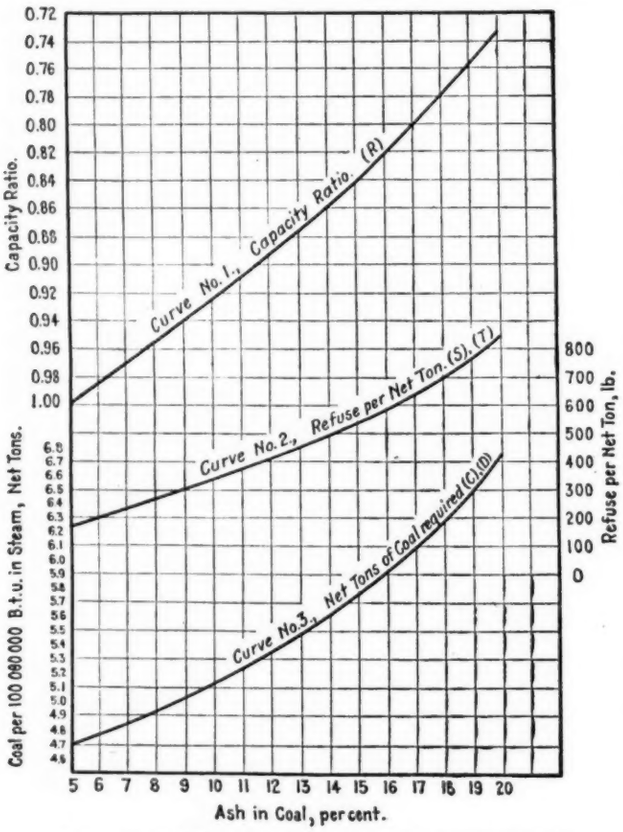
By E. B. RICKETS

THE costs of handling, storage and other steps in the process of obtaining heat are vitally influenced by the characteristics of the coal used. A market quotation represents only one of several quantities that constitute the true cost of coal.

The factors that determine the value of a certain coal in any given plant are as follows:

1. The cost in the bunkers of the coal required to produce a given number of British thermal units in the steam;
2. The relative investment in boiler plant, including coal- and ash-handling machinery, land, buildings, etc.;
3. The cost of handling and disposal of refuse;
4. The relative cost of fireroom labor with coals different in quality;
5. The cost of carrying banked boilers varying with the quality of the coal;
6. The clinkering tendencies of the coal.

The application of these factors is shown by curves and formulas, examples being worked out to illustrate specific


Capacity Ratio.
Coal per 100 000 000 B.t.u. in Steam, Net Tons.
Refuse per Net Ton, lb.
Curve No. 1, Capacity Ratio (R)
Curve No. 2, Refuse per Net Ton (S), (T)
Curve No. 3, Net Tons of Coal required (C) (D)
Ash in Coal, per cent.

RELATION OF COST FACTORS AND PERCENTAGE OF ASH IN COAL

Examples are given, showing the application of these curves and formulas to various plant conditions.

cases. The formulas are applicable to all types of coal-burning equipment. The curves, however, are based only on operating results with underfeed stokers.

The essential features of a coal contract are discussed and the fact emphasized that, if the contract contains a bonus and penalty clause, this bonus and penalty should equal the change in value of the coal to the consumer.

Having decided in this instance, that the coal supply is to be drawn from the semi-bituminous field, it remains to select that coal possessing the characteristics most desired. Experience in the operation of the plants under consideration has shown that the most satisfactory results can be obtained from coals having the following characteristics: (1) A volatile content less than 20 per cent; (2) as low ash as possible; (3) an ash fusion temperature not less than 2,450 deg. F.; (4) at least 25 per cent of the coal should be capable of being stored for nine months without serious trouble from spontaneous combustion.

The Samples Are Given a Boiler Test

Having selected from the coals offered those which, from consideration of price, quality, etc., seem to be most suitable, sample cargoes of these coals are obtained and subjected to a boiler test which weeds out a number of coals that laboratory tests may have indicated would be satisfactory.

We now have before us a number of coals offered at various prices, all of which our tests indicate will give satisfactory results in the boiler room. It still remains to determine their relative values.

All the six factors first enumerated, with the exception of the last one, bear a very definite relation to the percentage of ash in the coal, and an attempt has been made in the accompanying curves (Fig. 1) and formulas to give a method by means of which a comparison can be made of the value of a coal of any given analysis and that of a coal containing 5 per cent of ash. In this curve no account has been taken of the effect on coal value of extra fireroom labor (other than that involved in handling and disposal of ashes), extra cost of operating banked boilers or clinkering tendencies of the ash. The reason for this is that the first two of these items in most plants is a very small proportion of the total effect on coal value of changes in ash percentage, and in the case of the third item there is a lack of definite data applicable to other than a very restricted type of coal-burning equipment.

The curves given on this sheet are based on experimental data obtained at the plants of the New York Edison Co., and the Union Electric Light and Power Co., at Cincinnati, Ohio. Each of these plants is equipped with an underfeed type of stoker, the first being equipped with low-set boilers and the second having a modern type of setting. These curves are the average of the experimental data of the two plants and represent the conditions to be expected in the average underfeed stoker plant.

Two sets of formulas are given for applying these curves to all conditions; the first set takes into account the relative coal cost of a given number of British thermal units in the steam plus the cost of ash handling and removal, and the second set, in addition to the items considered in the first set, takes into account the cost of extra investment necessary in boiler plants to burn the higher-ash coal.

Formulas for Use in Calculating Coal Values

Three thousand boiler horsepower maximum rating will produce 100,000,000 B.t.u. in steam per hour. If the boiler plant, including land, buildings, etc., cost $100 per maximum horsepower when using 5 per cent ash coal, then such a plant would cost $300,000, which at 15 per cent (interest 7 per cent, depreciation 6 per cent, taxes and insurance, 2 per cent) equals $45,000 per year, or $5.13 per hour. Let

X = Cost in dollars per net ton of 5 per cent ash coal in bunkers (exclusive of ash handling);
Y = Value in dollars per net ton of x per cent ash coal in bunkers (exclusive of ash handling);
A = Load factor;
B = Cost of boiler plant per maximum horsepower divided by 100;
Z = Value in dollars per net ton of x per cent ash coal in bunkers;
C = Tons of 5 per cent ash coal to produce 100,000,000 B.t.u. in steam (4.67) (Curve No. 3);

*Abstract of paper read at (June 26, 1922) meeting of American Society for Testing Materials.

D = Tons of x per cent ash coal to produce 1,000,000,000 B.t.u. in steam (Curve No. 3);
R = Capacity Ratio[2] (Curve No. 1);
S = Pounds of refuse per ton from x per cent ash coal (Curve No. 2);
T = Pounds of refuse per ton from 5 per cent ash coal (160 lb.) (Curve No. 2);
U = Cost of ash handling and removal in dollars per pound.

Formulas, relative value neglecting investment:

$$Y = \frac{CX}{D} \quad (1)$$

$$Z = Y - (S - T)\ U \quad (2)$$

Formulas, relative value including investment:

$$Y = \frac{CX + \frac{5.13B}{A} - \frac{5.13B}{AR}}{D} \quad (3)$$

$$Z = Y - (S - T)\ U \quad (4)$$

Example 1, where investment cost is not considered: When 5 per cent ash coal costs $3 per net ton in bunkers, and ash handling and removal costs 20 cents per ton, to determine the economic limit of cost for coal having 20 per cent ash:

We have $X = 3$, $C = 4.67$, $D = 6.75$ (Curve No. 3), $S = 840$, $T = 160$ (Curve No. 2), and $U = 0.0001$.

Substituting in equations 1 and 2,

$$Y = \frac{4.67 \times 3}{6.75} = \frac{14.01}{6.75} = 2.08$$

$$Z = 2.08 - (840 - 160) \times 0.0001 = 2.01$$

Example 2, where investment cost is considered: When 5 per cent ash coal costs $3 per ton in the bunkers, ash handling and removal costs 20 cents per ton, the load factor is 50 per cent and additional boiler plant would cost $90 per maximum horsepower with 5 per cent ash coal, to determine the economic limit of cost for 20 per cent ash coal:

We have $X = \$3$, $A = 0.50$, $B = 0.90$, $C = 4.67$, $D = 6.75$ (from Curve No. 3), $R = 0.736$ (from Curve No. 1), $S = 840$, $T = 160$ (from Curve No. 2), $U = 0.0001$.

Substituting in equations 3 and 4,

$$Y = \frac{4.67 \times 3 + \frac{5.13 \times 0.90}{0.50} - \frac{5.13 \times 0.90}{0.50 \times 0.736}}{6.75}$$

$$= \frac{6.75}{10.71} = \$1.59$$

$$Z = 1.59 - [(840 - 160) \times 0.0001] = \$1.52$$

Starting Current Required for Induction Motors*

A study of the design of the standard induction motors of two of the leading manufacturers shows that for all sizes from 5 hp. up the locked-rotor starting current with full-line voltage applied is practically a straight line; that is, the amperes per horsepower are practically constant. For three-phase 220-volt motors this value is of the order of 15 amperes per phase per horsepower; that is, approximately 7½ times the running current of a theoretical unity power-factor 100 per cent efficient motor. The present N.E.L.A. rules permit for motors above 30 hp. a maximum line current of 6 amperes per phase per horsepower at 220 volts, which value is 75 per cent of the permissible locked-rotor value. The corresponding locked-rotor value is therefore 8 amperes, or about 50 per cent of that of the motor with full voltage applied.

[1]Load Factor = Total steam generated in pounds per year / Pounds of steam generated in maximum hours × 8,750.

[2]Capacity Ratio = Maximum capacity of boiler plant with 20 per cent ash coal / Maximum capacity of boiler plant with 5 per cent ash coal.

*Abstract from N. E. L. A. Electrical Apparatus Committee's, 1922, Report.

Since the line current decreases as the square of the voltage applied to the motor, a starting compensator connected to supply 70 per cent of line voltage to the motor will limit the starting current to that required by the rules. For very large motors, 500 hp. and up, this is deemed excessive by some operators, who would like to limit the line starting current to 4 instead of 6 amperes per phase per horsepower, or to about twice full-load current. This would require an applied motor voltage of 58 per cent of line voltage. As the starting torque of a motor also decreases as the square of the applied voltage, it follows that only such motors whose starting duty is light could be connected to such low starting taps.

Effect of Starting Compensators

The effect of starting compensators may be summed up in the following tabulation:

Standard Taps In Per Cent of Full Voltage	Size of Starter	Corresponding Line Current in per Cent Of Line Current At Full Voltage	Corresponding Torque in per Cent Of Starting Torque At Full Voltage
40	5 to 18 hp.	16	16
58	5 to 18 hp.	34	34
70	5 to 18 hp.	50	50
85	5 to 18 hp.	72	72
50	19 hp. and larger	25	25
65	19 hp. and larger	42	42
80	19 hp. and larger	65	65
100	All sizes running position	100	100

The minimum starting torque in percentage of full-load torque with full voltage applied varies from 115 per cent for slow-speed motors to 150 per cent for high-speed motors. The maximum running torque varies from 175 per cent to 250 per cent of full-load torque.

A machine requiring a torque of 120 lb. at one foot radius to start it and a torque of 250 lb. at its heaviest duty period, could be driven by a 25-hp. motor having the following characteristics: Full-load torque, 146 lb. at one foot radius; starting torque, 180 lb. at one foot radius; maximum running torque, 325 lb. at one foot radius.

The starting torque required is 56 per cent of that of the motor with full voltage applied. If connected to the 70 per cent tap of a starting compensator, the motor would develop a torque of 90 lb. (50 per cent of 180 lb.) and therefore could not start the load. By connecting to the next higher standard tap (85 per cent) a starting torque of 72 per cent of 180 lb., or 130 lb., would be developed, or sufficient to start the load easily. The starting current would, however, be 72 per cent of 15 amperes, or approximately 11 amperes per phase per horsepower, assuming a 220-volt three-phase motor.

Rigid enforcement of the suggestion that line starting currents be limited to twice full-load current would in many cases result in compelling the application of larger motors to the load in order to obtain the necessary starting torque with all the evils consequent to over-motoring.

Going back to the illustration to limit the starting current to 4 amperes on the line, a starting tap of 50 per cent would be required. Since the starting torque under these conditions is also 25 per cent of that of the motor with full-voltage applied, a motor having a starting torque of 4 times 120 lb., or 480 lb., would be required. The nearest standard size would probably be a 50-hp. motor. If, on the average, the 25-hp. motor originally selected were loaded to 80 per cent of its capacity, it would operate at about 82 per cent power factor. The 50-hp. motor required to keep the starting current down, would be loaded to 40 per cent of its capacity and would operate at about 75 per cent power factor.

With this in mind the two larger manufacturers were asked what would be the effect of designing motors with locked-rotor starting-current values of two-thirds of the present design values. The replies stated that in order to accomplish this the reactance would have to be increased, thus reducing the power factor about 3 per cent. The resistance would be unchanged in order not to reduce present efficiencies. The starting torque would be reduced to about four-ninths of the present value and the maximum output torque to about two-thirds of the present value. The manufacturers feel that these torque reductions would be such as to make motors designed in this manner commercially undesirable.

Secretary Merrill Boosts Water Power for Southern Appalachian States

That the five southern Appalachian states of North and South Carolina, Georgia, Alabama and Tennessee have made particularly rapid progress in developing their water-power resources and that they should make plans for continuing such development to the maximum, were the two main points of an address made recently in Asheville, N. C., by O. C. Merrill, executive secretary of the Federal Power Commission. He pointed out that the central stations in these states had produced in 1907 only 3½ per cent of the total central-station energy in the United States, while in 1917 they were supplying nearly 8 per cent, the output for 1917 having increased to 8½ times that of 1907. There has been a corresponding increase in installation, he said, and while less than half of this installation was water power in 1907, it was nearly two-thirds in 1917, a ratio approached by only one other section, the far West. "The states represented here have, however," said Mr. Merrill, "made only a beginning in the use of their potential water power. The time is now here when you should secure the necessary information and should draw the plans for the fullest utilization of the resources which you have."

In explaining the matter, Mr. Merrill gave a review of the general power situation, including some interesting figures. He illustrated the tremendous size and the rapid growth of the power field by pointing out that in 1907 the total output of central stations in the United States was two billion kilowatt-hours per year, in 1917 it was twenty-five billion, and in 1920 forty-four billion. He explained, however, that these figures represent only electricity, without taking into account power that is produced and used directly; he estimated that if all our power needs—domestic, industrial and transportation—were served electrically, our annual requirements would probably amount to one hundred and fifty billion kilowatt-hours, thus showing that electric energy has thus far entered less than one-third of the power field.

The present total of installed power in manufacturing plants, central stations, railways, etc., was estimated at from seventy to eighty million horsepower, of which water power is estimated to be not over nine million horsepower, or a little over 10 per cent. Although recognizing the probability that the energy output of hydro plants per horsepower installed is somewhat greater than for fuel plants, Mr. Merrill said that we can hardly assume that more than 15 per cent of our total power requirements are met by water power.

The speaker pointed out that the proportionately small amount of water power that is developed is not due to any lack of such resources, inasmuch as we have far more of them than we can use now, nor is it primarily due, he said, to unsuitable legislation, although the latter has been a contributory cause. He considered the two main reasons for the situation to be these: First, since the transmission of electric power over long distances and the application of such power to industrial needs are developments of comparatively recent years, our industries have been organized on a basis of steam or other direct power, and the change from that to water power involves considerable expense, and is also held back by inertia; second, geographical conditions are unfavorable, because most of our industries are in the East and most of our water power is in the West.

In illustrating the latter point, he showed that over three-fourths of the prime-mover capacity in the United States used in manufacture, in central stations and in electric railways, is east of the Mississippi, and nearly two-thirds east of the Mississippi and north of the Ohio and the Potomac; in this area, also, he said, is the greater part of our railway traffic. Right at hand, in this same district, he continued, in the six states of Pennsylvania, Ohio, Indiana, Illinois, West Virginia and Kentucky, 80 per cent of our bituminous coal is mined. Of our estimated water-power resources, however, he said that three-fourths are west of the Mississippi and 60 per cent are in the five states of California, Oregon, Washington, Idaho and Montana.

The Future Development of the Marine Diesel Engine

In a paper read before the Diesel Engine Users' Association, H. F. P. Purday discussed the marine Diesel engine. In referring to the main points of difference in the design of the marine as compared with the land engine, he pointed out the fact that the hull of the ship, being subject to more or less violent motion, did not form the rigid foundation which was furnished in the case of the land engine. It was therefore necessary to provide for considerable rigidity in the engine frame or in the seating in order to avert troubles due to the main bearings getting out of alignment. Crankshafts must be considerably stronger than those used in land practice. To provide for ability to start up from any crank position, it was also necessary to use engines having a certain minimum number of cylinders—at least six for the four-stroke-cycle engines and four for the two-stroke-cycle engines.

After referring to the principal types of Diesel engines, the author dealt with the subject of the four-stroke-cycle and the two-stroke-cycle engines and referred to opposed piston two-stroke-cycle engines using solid injection and air injection of fuel. The Still engine was mentioned as being in a class by itself. There was considerable controversy on the subject of heat stresses, and there could be no question that the provision of relatively small two-stroke cylinders to perform the same work as relatively large four-stroke cylinders did entail some thermal problems of an important character.

In dealing with fuel-injection systems, Mr. Purday stated that so far the best fuel consumption with the solid-injection system seemed to be associated with high maximum pressures, which were penalized by the registration societies. When the maximum pressure was limited to 500 lb. per sq.in., the fuel consumption was usually slightly higher than with the air-blast system.

The permissible mean pressure for two-stroke-cycle engines depended largely on the efficiency of the scavenge process, while the piston speeds were generally between about 500 and 700 ft. per min. Comparatively slow piston speeds had been adopted on account of the desirability of making use of low scavenge pressure in order to secure a good mechanical efficiency and to keep the revolutions low on account of the propeller efficiency. Two-stroke-cycle engines, however, were not necessarily restricted to low piston speeds.

The author discussed the thermal difficulties of large Diesel engines and pointed out that the trend of modern design seemed to be to eliminate extensive temperature stresses by dividing the castings into smaller pieces having considerable freedom from expansion. He favored the practice of providing concave crowns to pistons as the farther the center of the crown was away from the fuel valve, the less heat it would presumably receive.

The view was expressed that the Diesel engine must be seriously considered in the near future for propelling large vessels requiring something like 16,000 shaft horsepower per shaft.

In the discussion that followed, the belief was expressed that further development of the marine Diesel egine was likely to follow along the lines of the double-acting two-stroke engine. Further interesting information was given concerning the cylinder construction of the "Still" engine. Reference was made to the greater use of the larger sizes of engine for land use on the continent and in other countries abroad. Stress was laid on the importance of reducing as far as possible the cost of manufacture and a point was made that the present high cost of the larger units for land use was due to the comparatively small demand up to date.

Congress, in the naval appropriation bill for the fiscal year beginning July 1, limits the fuel appropriation for the Navy during that year to $16,000,000, notwithstanding requests of the Navy backed personally by Secretary Denby that $19,894,000 be given.

News in the Field of Power

Duquesne Company Planning Big Heating Plant

One of the largest heating plants in the country is to be put up by the Duquesne Light Co., of Pittsburgh, according to present plans, which call for completion of the plant by the fall of 1923. The plant is to be capable of generating a million pounds of steam, so that the total heating surface of the boilers to be installed will be some 300,000 sq.ft.

It is expected that the building will have to be put up on limited ground area and that it will be about 150 ft. high.

Navy Saves with Oil Purifiers

Admiral Robison, chief of the Navy Bureau of Engineering, in testimony before the Senate appropriations committee in behalf of funds for his service during the coming year, referred to water getting into oil in oil-burning ships, leaving the ships without power. To overcome this difficulty, electric feed pumps and fuel-oil service pumps running from a storage battery are required. He recommended fuel-oil refiners for the ships, stating that the Navy has placed oil purifiers on every battleship and destroyer to purify lubricating oil, and that these devices are saving an average of their full cost every year.

Boiler Fees Now Graduated

C. O. Meyers, secretary-treasurer of the National Board of Boiler and Pressure Vessel Inspectors, has announced that the basis of charging fees for registering boilers has been changed from the fixed charge of $2 per boiler to a sliding scale depending upon the heating surface of the boiler. The following schedule has been approved, effective July 1, 1922:

From 1 to 5 hp., $0.25; 6 to 50 hp., $0.50; 51 to 200 hp., $1; 201 to 500 hp., $2; 501 hp. up, $3.

In this table 1 hp. is, of course, taken as representing 10 sq.ft. of heating surface.

It is pointed out that it will now be necessary for manufacturers to place the heating surface of the boiler at the bottom of the data report.

An Intensive Course in Industrial Administration

From Aug. 28 to Sept. 9, at Pennsylvania State College, there will be held the Seventh Annual Summer Session in Industrial Organization and Administration, under the immediate direction of Prof. Edward J. Kunze, assisted by Prof. J. O. Keller and P. P. Henshall, of the Department of Industrial Engineering. The course is to be an intensive one, designed to show the most effective methods of modern organization to manufacturers, superintendents, personnel directors, accountants, production managers and other executives.

The greater part of the time is to be given over to practical installation methods, using the equipment of the five shops and other facilities of the school. There will also be group conferences, a feature that has proved valuable before. The different lines of manufacture represented by those who will attend the course is expected to be very wide. Bulletins about the course may be had by addressing Professor Kunze at State College, Pa.

President Harding, to Coal Operators and Miners—

"The government has no desire to intrude itself into the field of your activities. It does feel an obligation to see that the common American interest shall not be menaced by a protracted lack of fuel. It prefers that the two great and associated interests—mineworkers and employers—should settle this matter in a frank recognition of the mutuality of your interests. If you cannot do that, then the larger public interest must be asserted in the name of the people, where the common good is the first and highest concern."

—An excerpt from a recent speech of the President, plainly setting forth the Government's attitude on the coal strike.

Boston Societies Affiliate

The engineering and technical associations of Boston have finally completed organization plans that have been under way for some time, and have joined as the Affiliated Technical Societies of Boston, with headquarters at 715 Tremont Temple, where the Boston Society of Civil Engineers and the New England Waterworks Association formerly had their offices. Nine societies are included, and their total "non-duplicating" membership is about 3,500. The affiliated organization has been granted a charter by the State of Massachusetts.

In general, the objects of the affiliation are to eliminate duplication in the programs given by the various member associations and to promote a stronger engineering feeling by bringing together all engineers in the vicinity. The officers of the affiliation council are: Chairman, Leonard Metcalf; vice-chairmen, Alfred S. Kellogg and Alexander Macomber; treasurer, Charles L. Hammond; clerk, W. G. Starkweather; executive secretary, J. B. Babcock.

Pennsylvania Will Not Allow Marine Boilers

A recent announcement from Commissioner Connelley, of Pennsylvania, is to the effect that boilers of the marine type, such as were built for the United States Shipping Board at Hog Island, are not eligible for installation in Pennsylvania. He points out that such boilers are not covered by the Code, and that "only boilers that were built for stationary purposes and in strict accordance with the Code can be installed and operated in the industrial establishments in the commonwealth." Several installations of unacceptable boilers have been reported, and for that reason the recent decision has been sent to all approved boiler inspectors in the State, with the warning that commissions of competency will be revoked in case of failure to comply with the regulations.

A Safety Code for Conveyors and Conveying Machinery

The American Engineering Standards Committee has decided to undertake the development of a safety code for conveyors and conveying machinery, and has announced the appointment of the American Society of Mechanical Engineers and the National Bureau of Casualty and Surety Underwriters as sponsors for the code. It is pointed out that although the introduction of mechanical conveyors has eliminated a large proportion of accidents that used to happen in handling materials manually, there are now some new dangers in the operation of the conveying machinery itself, and it is held that these dangers can be lessened by a more general agreement among both users and manufacturers of conveying equipment as to safe practices in the manufacture, installation and operation of such apparatus.

The code will cover all types of conveyors, including gravity, belt, chain, aërial cableways, etc. Shafting, pulleys, belts, gears, clutches, etc., used in connection with such machinery, however, will be covered by the mechanical power transmission code, which is now in preparation.

All interested national organizations will be asked to co-operate in preparing the code, and suggestions from all responsible sources will be welcomed by the committee, whose headquarters are at 29 West 39th St., New York City.

The suit of New York against the constitutionality of the Federal Water Power Act in its application to State streams was not reached by the Supreme Court at its term recently closed. This case will probably be argued early in the new term beginning in October.

New Publications

Methods for Testing Petroleum Products. Technical Paper 298 of the United States Bureau of Mines, Washington, D. C. Limited free edition.

Those who are responsible for the buying of considerable quantities of petroleum products will do well to obtain a copy of this pamphlet, which describes the methods of testing used by the Government in buying petroleum for its own uses. These methods were adopted by the Interdepartmental Petroleum Specifications Committee, and supersede similar methods published in Bulletins 1 to 5 inclusive of the Committee on Standardization of Petroleum Specifications.

The Four-Hour Day in Coal. By Hugh Archibald. Published by the H. W. Wilson Co., 958-64 University Ave., New York City. Cloth; 5 x 8 in., 148 pages; illustrated, Price $1.50.

A timely little book. If everybody would read it, there would be a more intelligent public appreciation of the conditions that lie at the root of the present trouble. It deals with the work and the conditions of the miner, not in the sentimental but in the engineering mood, devotes a chapter to grievances and then tells what coal mining is like, treats of the mine foreman and his problem; time for the miner; amount of work for the miner; record of complaints in Illinois; discouragements and the lack of thrift and the engineering need.

The author writes in the light of twenty years' experience in and about the mines, and presents the facts upon which the engineering mind can build the systematic administration of our most important resource.

Steam Heating.—A manual of practical data, compiled by the General Engineering Committee of Warren Webster & Co., published by Warren Webster & Co., Camden, N. J. Cloth; 8 x 10½ in.; 367 pages; 277 illustrations and 175 tables. Price, $3.25 net.

Any large company that specializes in a given field necessarily acquires a vast fund of practical information. The old habit of hoarding such information is gradually dying out as the advantages of sharing it become more evident, and in recent years several books of high engineering value have been published by manufacturing concerns. The present volume gives an immense amount of data of the sort that could be obtained only by a broad experience in the field of heating. Some sections, such as those dealing with the heating of residences, will not be of interest to the power-plant engineer, but most of the book has a direct bearing on the daily work of any engineer who is in charge of an industrial heating system.

The first 150 pages deal with the subject of heating in general, with unusually complete data on the following subdivisions: Data required for steam-heating system designs; heat transmission; air infiltration; method of calculating heat requirements; method of computing and selecting heating surface; ventilation problems as they affect the design of heating systems; proportioning of chimneys; boilers; selection of the proper type of steam-heating system; flow of low-pressure steam through piping; critical velocities in radiator runouts; vacuum pumps and auxiliary equipment; laboratory tests of return traps. The second section of the book deals with the specialties used in the Webster systems of steam heating, while the last 40 pages are given over to useful miscellaneous information of interest to engineers. It is to be hoped that the example set by the Webster company will be followed by manufacturers in other fields, where the preparation of comprehensive treatises would be of great value to the engineering fraternity.

Mexican Year Book, 1920-1921. By Robert Glass Cleland, Ph.D. Published by Mexican Year Book Publishing Co., Los Angeles, Cal. Cloth; 6 x 9 in.; 524 pages. Price, $7.50.

The announced object of the book is to meet the demand for unbiased, systematized information about Mexico, prepared for the American public. The preface compares it with preceding Mexican Year Books as being better adapted for American use and as having no connection, official or unofficial, with the Mexican government. The subjects covered are: History and geography, politics and government, travel and transportation, commerce and manufacturing, natural resources, public finances, currency and banking, labor conditions and educational systems. A bibliography of works on Mexico, suggestions for the business traveler in Mexico and similar information are also included.

Notes and Examples on the Theory of Heat and Heat Engines. By John Case, M. A. Published by W. Heffer & Sons, Ltd., Cambridge, England. Handled in United States by D. Van Nostrand Co., 8 Warren St., New York City. Cloth; 5½ x 8½ in.; 136 pages; 64 numerical problems. Price, $2.50 net.

If this book does not meet the needs of the average power-plant engineer, no blame should attach to the author, who evidently had quite a different purpose in mind. In the words of the author, "The book is not intended as an exhaustive textbook, as sufficient excellant treatises exist already, but rather a companion to lectures, to help the student to see at a glance the important points of the subject and to assist him with his revision for examinations." In other words, the book is primarily for the assistance of students who are hearing lectures on thermodynamics in British universities.

The method of treatment is mathematical throughout and involves a knowledge of calculus and differential equations, making the book largely unintelligible to the average engineer. From the point of view of American practice, the book has another bad feature, in that it employs what is called the standard thermal unit, which is defined as the amount of heat required to raise 1 lb. of water 1 deg. C. In the United States the British thermal unit is used almost exclusively by engineers and the metric unit, the calorie, by workers in pure science. The introduction of another unit, half metric and half English, would seem to be an unnecessary complication in a field that is already far more complicated than is desirable.

The Ignition Temperature of Coal. Bulletin No. 128. By Ray W. Arms. Published by the Engineering Experiment Station, University of Illinois, Urbana, Ill. Paper, 6 x 9 in., 70 pages. Price, 35 cents.

Ignition temperature has been defined by those who carried on the work described in the book as the temperature at which coal, when heated slowly, begins to glow. No fixed relation has been discovered between this point and the liability of any particular coal to spontaneous combustion, but the data given may nevertheless be of some value to those who make a particularly keen study of the combustion of coal. The booklet covers a narrow subject, but does so thoroughly. Numerous other valuable publications of a similar nature are put out by the Engineering Experiment Station, most of them being obtainable at a small charge.

Personals

Charles T. Main, widely known consulting engineer, on July 1 moved his organization from 201 to 200 Devonshire St., Boston, Mass.

R. B. Mildon has been appointed general manager of the stoker department of the Westinghouse Electric and Manufacturing Co., with headquarters at South Philadelphia.

C. W. Fields, for four years an engineer with the Carnation Milk Products Co., at Kent and Mt. Vernon, Wash., is now working at installation and "trouble shooting" with the Edwards Ice Machine Co. at Seattle.

L. E. Armstrong has taken a position as mechanical engineer with the Bass Foundry & Machine Company of Fort Wayne, Ind. He was formerly in the engineering department of the Heine Boiler Company.

Anson W. Burchard has been elected president and chairman of the board of the International General Electric Company. He was formerly vice-chairman of the board of directors, and succeeds Gerard Swope, who recently became president of the General Electric Company, and Charles Neave, who resigned.

Royal D. Holbrook of Cedar Rapids, Iowa, in addition to his work as combustion engineer for the Department of Engineering Extension of Iowa State College, has been appointed traveling

engineer for the State Board of Control, which has under its jurisdiction sixteen state institutions burning about $750,000 worth of fuel per annum. He is being prominently mentioned as a candidate for the vice-presidency of the N.A.S.E. at the coming convention.

Society Affairs

Coming Conventions

National Association of Stationary Engineers; Fred W. Raven, 417 South Dearborn St., Chicago, Ill. Annual conventions and exhibitions of the State Associations scheduled as follows:

New England, at Springfield, Mass., July 12-14; James Morgan, 53 Devonshire St., Boston.

Michigan, at Kalamazoo, July 19-21; A. M. Adams, 1307 Stockbridge Ave., Kalamazoo, Mich.

Wisconsin, at Madison, July 26-28; C. Z. Wise, 204 South Carroll St., Madison.

Minnesota, at Minneapolis, Aug. 9-11; C. A. Nelsen, 800 22nd Ave., N. E., Minneapolis.

Iowa, at Des Moines, Sept. 10-11; Abner Davis, Room 16, Waterhouse Block, Cedar Rapids, Iowa.

Universal Craftsmen Council of Engineers; Thomas H. Jones, 33 Linden Ave., Cherrydale, Va. Annual convention and exhibition at Cleveland, Aug. 7-11.

The National Association of Practical Refrigerating Engineers has set the date for its Thirteenth Annual Convention and Exhibition as Nov. 1-4, at the Planters Hotel in St. Louis. The entire second floor of the hotel has been engaged for exhibition and meeting purposes. The objects of the Association are educational, and its members are chief and operating refrigerating engineers. The secretary is Edward H. Fox, 5707 West Lake St., Chicago.

The Engineers' Society of Milwaukee has announced the election of the following officers for the ensuing year: President, Arthur Simon, engineer of the Cutler-Hammer Manufacturing Co.; vice-president, Fred H. Dorner, mechanical engineer; treasurer, A. Q. Dufour, mechanical engineer; secretary, Fred T. Goes, engineer of the Vilter Manufacturing Co. At the conclusion of the annual meeting, Fred H. Dorner was presented with a gold watch in appreciation of his work as secretary for the last ten years..

Business Items

The Chicago Pneumatic Tool Co., 6 East 44th St., New York City, has made the announcement that, on June 27, Charles M. Schwab was elected chairman of the board of directors, succeeding John R. McGinley, who has resigned.

The Westinghouse Electric and Manufacturing Co. has made several important changes in its power sales department. L. C. Bullington is now to have charge of the general work of the power department, having been appointed assistant manager. Charles F. Lloyd, formerly manager of the substation section, is now manager of the electric division, and R. E. Carothers, formerly manager of the turbine section, is now manager of the steam division. Mr. Lloyd and Mr. Carothers have been succeeded by Bruce H. Lytle and D. O. Tylee.

The Economy Fuse and Manufacturing Co. has moved its Pittsburgh sales office from the Farmers' Bank Building to the People's Bank Building, at Fourth Ave. and Wood St.

The New Haven Boiler Works, Inc., New Haven, Conn., has announced the election of Eugene L. Cooke as secretary and treasurer. Mr. Cooke was formerly with the Vincent-Gilson Engineering Co., Inc., of New York City.

The Pittsburgh Valve, Foundry and Construction Co., Pittsburgh, Pa., has moved its Chicago sales office from the McCormick Building to the Monadnock Block. J. B. Longini is district representative.

The United Machine and Manufacturing Co., Canton, Ohio, has appointed the Hale-Stephen Co., with offices in Cleveland and Detroit, as its sales representative for the Harrington traveling-grate stoker and the industrial underfeed stoker.

Wilson Welder and Metals Co., 132 King St., New York City, has appointed the King-Knight Co., Underwood Bldg., San Francisco, Cal., as exclusive representative for the sale of plastic-arc welding machines and "color-tipt" metals in the central and northern California districts.

Trade Catalogs

Flexible Couplings—Thomas Flexible Coupling Co., Warren, Pa. A short catalog describing the Thomas flexible coupling and containing some unusually interesting photographs showing a 10,000-hp. coupling installed in a steel mill.

Corliss Engines—Chuse Engine and Manufacturing Co., Mattoon, Ill. Bulletin No. 7, 24 pages, describing a line of single, tandem and cross-compound non-releasing Corliss engines designed primarily for direct connection to electric generators.

Turbines and Alternators — Allis-Chalmers Manufacturing Co., Milwaukee, Wis. Bulletin No. 1122, 54 pages, describing Allis-Chalmers steam turbine and generator units of the non-condensing and condensing types, with pressure-operated bypass valve. Interesting detail photographs are included, as well as a number of drawings, showing clearly the construction of the apparatus.

Fuel Oil Burning Systems—Schutte & Koerting Co., Philadelphia, Pa. Vol. 5, containing five well-prepared and interesting bulletins on "Koerting Mechanical Oil Burning System," "Auxiliary Apparatus for Burning Oil," "Koerting Asko Type Oil Burning System," "Koerting Steam and Air Pressure Oil Burning System," and "Fuel Oil, Its Properties and Advantages."

Grates and Hand Stokers—Auburn Foundry Corporation, Auburn, Ind. A new 20-page catalog on the Auburn "Clear-Cut" grate and the Auburn hand stoker.

Operating CO_2 and Draft Recorder—Jos. W. Hays Corporation, Michigan City, Ind. A 32-page book of operating instructions for the Hays Automatic CO_2 and draft recorder. The explanation of the principles of such apparatus has been prepared with unusual care and thoroughness and should be of great value to those desiring an understanding of such equipment.

Heating and Temperature Regulating Devices—The Fulton Co., Knoxville, Tenn. Catalog No. 100, the first general catalog published by this company, which has recently been reorganized. The booklet contains 145 pages describing "Sylphon" heating specialties, such as temperature and pressure-controlling instruments, valves, etc. Many useful tables and diagrams are also included.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine-run bases, f.o.b. mines):

Coal	Market Quoting	June 26, 1922	July 3, 1922
Pool 1,	New York	$4.40–4.50	$4.75–4.85
Smokeless,	Columbus	3.35–3.50	3.35–3.50
Clearfield,	Boston	3.15–3.40	6.10–6.25
Somerset,	Boston	3.35–3.50	3.35–3.60
Kanawha,	Columbus	3.25–3.50	3.25–3.50
Hocking,	Columbus	3.40–3.50	3.25–3.50
Pittsburgh No. 8	Cleveland	3.75–4.00	3.90–4.10
Franklin, Ill.,	Chicago	.....	.....
Central, Ill.,	Chicago	.....	.....
Ind. 4th Vein,	Chicago	.....	.....
West Ky.,	Louisville	4.00–4.15	4.25–4.50
Big Seam,	Birmingham	1.85–2.00	2.00–2.25
S. E. Ky.,	Louisville	3.25–3.50	3.50

FUEL OIL

New York—July 6, Port Arthur light oil 22@25 deg. Baumé 3¼c. per gal.; 30@35 deg., 4¾c. per gal. f.o.b. Bayonne, N. J.

Chicago—June 28, for 24@26 deg. Baumé, $1.15 per bbl.; 21@36 deg., 3½c. per gal. in tank cars, f.o.b. Oklahoma refinery, or freight adjusted.

Pittsburgh—July 5, f.o.b. refinery, Pennsylvania, 36@40 deg., 5¼c.; Kentucky fuel oil, 32@34 deg., 4½c. per gal. Gas oil, 32@34 deg., 3c. per gal.; 36@38 deg., 3¼c.; 38@40 deg., 3½c. Western, 24@30 deg., $1.35 per bbl.

Philadelphia—July 3, 26@28 deg. Baumé, Oklahoma, $1.05@$1.10 per bbl.; 30@34 deg., Oklahoma (group 3), 3@3¼c. per gal.; 16@20 deg. Seaboard, $1.15@$1.25 per bbl.

Cincinnati—May 27, for 26@30 deg. Baumé, 5½c.; Diesel 32@24 deg., 5½c. per gal.; distillate 39@40 deg., 6c.

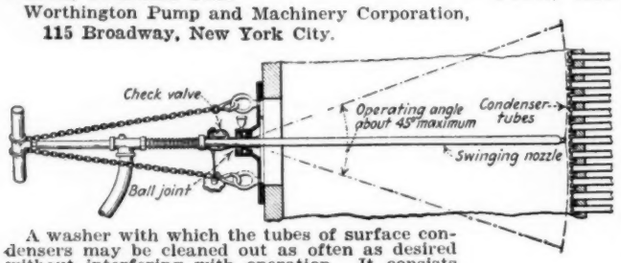
Cleveland—May 27, 26@30 deg. Baumé, 4¾c. per gal.

Condensed-Clipping Index of Equipment

Condensed from previous descriptions in "Power" for convenient filing

Washer, Condenser Tube "Power," 1922.

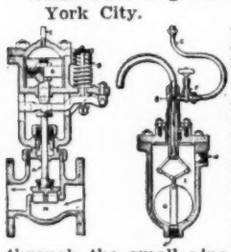
Worthington Pump and Machinery Corporation, 115 Broadway, New York City.



A washer with which the tubes of surface condensers may be cleaned out as often as desired without interfering with operation. It consists of a water-jet tube that is inserted through attachments on the various manhole plates and swiveled around so as to force a jet of water (about 60 gal. per min. at 250 lb. pressure) through each tube in succession. When the water jet is withdrawn, a check valve closes and prevents the escape of water. A cap is then screwed on to seal the joint. Further described in "Power" for May 9, page 732.

Governor, Feed Pump "Power," 1922.

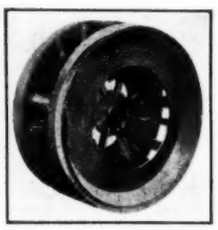
White Fuel Engineering Corporation, 742 West 12th St., New York City.



The complete set of governing apparatus consists of three pieces: A float-operated steam pilot valve that is connected with the water and steam spaces of the boiler; a steam-operated valve (controlled by the float valve) placed in the feed line between the pump and the boiler; a hydraulically-operated valve, which is the governor proper, for controlling the speed of the feed pump. The first two are shown here, but the governor is omitted for lack of space. The operation is as follows: The water level in the float valve (shown at the right) is the same as that in the boiler, and when it rises the float also rises and lets steam through the small pipe *C*, which is connected to the steam-line valve (shown at the left) at the top, where it is also marked *C*. The steam then forces down the piston *B*, closing the valve *H* and shutting off the water supply to the boiler. The resulting increase in pressure on the discharge side of the pump operates a pilot valve in the pump governor. Further described in "Power" for Jan. 31, page 184.

Fan, Cinder Catching "Power," 1922.

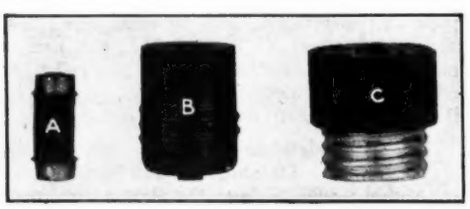
B. F. Sturtevant Co., Boston, Mass.



An induced-draft fan designed with the double purpose of causing a draft and of separating cinders and finer particles of solid matter ordinarily carried out by the flue gases and dropped on the surrounding neighborhood. A number of guiding vanes are provided on the blades of the fan to throw the cinders into a dust chamber, from which they settle by gravity into a hopper at the bottom. The cinders can then be removed, from time to time, or continuously by screw or other conveyors. It is claimed to be more efficient than the ordinary induced-draft fan and to remove 75 per cent of the solid matter in the flue gases. It is said that a test made on two of these fans installed with six 5,000 sq.ft. boilers resulted in the separation by each fan of 250 lb. of cinders an hour, and that the separated material resembled a fine coke breeze that showed upon analysis about 9,700 B.t.u. per pound. Described further on p. 180 of "Power" for Jan. 31.

Fuse, Refillable, Plug Type "Power," 1922.

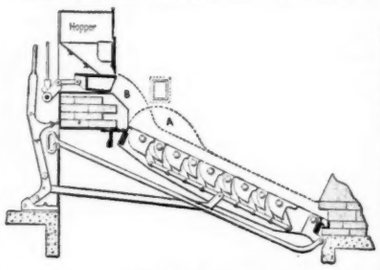
Cote Bros. Manufacturing Corporation, Chicago, Ill.



A simple refillable plug fuse, consisting of only three parts, marked *A*, *B* and *C* in the illustration. *A* is a refillable cartridge, or small cartridge fuse; *B* is the body, made of a heat-resisting molded insulation and containing the bottom contact of the fuse; *C* is the cap, made of molded insulation, threaded on the inside to receive the corrugated brass part. This brass part screws over some threads on the body *B* and also screws into the socket; as it is closed at the upper end, it makes contact with the fuse proper and so closes the circuit. There is a small hole through the center of the brass part, and also through the top of the cap, through which can be seen the amperage. Described further on p. 177 of "Power" for Jan. 31.

Stoker, Hand, Hopper Feed Added "Power," 1922.

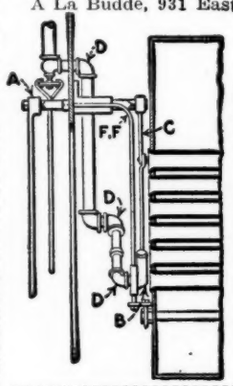
Cokal Stoker Corporation, 341 East Ohio St., Chicago



A hopper feed has been added to the standard Cokal hand stoker, thus eliminating hand firing and making it possible to receive the coal directly from conveying equipment or overhead bunkers. This prevents the excess air that would come in through the firing doors, and also tends to maintain automatically the correct shape of fuel bed, as indicated at *B* in the illustration. With hand firing, a careless operator sometimes fires in such a way that a pile of fresh coal is carried in the front part of the stoker, as indicated at *A*, thus reducing the active area of the fuel bed. With the hopper-feed arrangement the coal is pushed into the furnace by the hand-operated feeder block. Further described in "Power" for May 16, page 774.

Blower, Flue, for H. R. T. Boilers "Power," 1922.

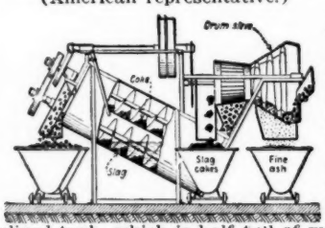
A La Budde, 931 East Third St., Duluth, Minn.



May be operated from the floor in front of the boiler by a hand lever on the squared shaft *A* and an extension rod on the steam-supply valve. Turning the shaft *A* raises the blower pipe *B*, through the lever *C*, thus blowing out first one row of tubes and then another. Steam nozzles are screwed into the blower pipe opposite each tube. The blower pipe is guided up and down by short pieces of pipe, which are spot-welded to it and which slide along the vertical pipes *F F*; it is supplied with steam through a pipe that is jointed at *D D D*. The device is made to order, according to the boiler on which it is to be installed. Described further in "Power" for April 25, page 643.

Separator, Coke "Power," 1922.

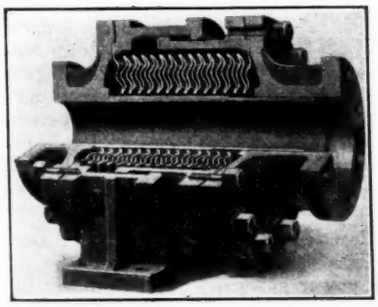
R. Stutzke Co., Inc., 526 West 18th St., Chicago, Ill. (American representative.)



A machine for recovering the combustible matter remaining in furnace refuse. It was first developed in Germany, and is now being introduced in this country. The furnace residue is dumped into the revolving drum sieve, from which the fine cinders and ashes sift through into the first dump car; large slag cakes pass through into a second car, and the remainder falls into one end of an inclined tank, which is half full of water that has been treated with clay or other material to give it a specific gravity of 25 to 30 deg. Baumé. The coke, being light, rises to the surface of the water, and the slag, being heavy, falls to the bottom. The two conveying screws deliver the coke and the slag to separate dump cars. The machine is made in three sizes, yielding from 1¼ to 12 cu.yd. per hour, requiring from 1 to 4 hp. The two smaller sizes are portable. Described further in "Power" for June 6, page 899.

Joint, Expansion "Power," 1922.

Ray Expansion Joint Co., Inc., 95 Liberty St., New York City.



An expansion joint for piping systems, consisting of two flanges, one of which may slide upon a tube that is tight in the other. This sliding joint, however, allows some steam leakage, which is kept from escaping by the peculiar expansion element shown in section in the illustration. This element is made up of a series of steel discs, which are welded together, each disc being fastened at its outside edge to one disc, and at its inside edge to another. The outer casing not only protects the discs from outside injury, but also keeps them from being stretched beyond their elastic limit. Described further on page 141 of "Power" for Jan. 24.

For a permanent file of new and improved power-plant equipment, clip and paste on 3 x 5-in. cards. (Patented Aug. 20, 1918.)

PRICES - MATERIALS - SUPPLIES

Electrical prices on following page are prices to the power plant by jobbers in the larger buying centers east of the Mississippi. Elsewhere the prices will be modified by increased freight charges and by local conditions.

SINCE LAST MONTH

Advances—Structural and boiler rivets up 10c in New York warehouses and 15c. per 100 lb, at Pittsburgh mill. Cold finished steel advanced 10c. in New York and 30c. per 100 lb. in Cleveland. Silica and magnesite brick up $3 and clay brick, $1 per M, during month. Linseed oil prices tending upward.

Declines—Cotton waste, white, down ¾c. and mixed, 1c. per lb. in Chicago. Babbitt metal, all grades, reduced 4½c. @11c. per lb. in New York warehouses, slightly lower in Cleveland.

POWER-PLANT SUPPLIES

HOSE—

	Fire	50-Ft. Lengths
Underwriters' 2½-in., coupled		47c. per ft.
Common, 2½-in., 3-ply		1.00 per ft. list less 50%

	Air	
	First Grade	Second Grade
¾-in., 3-ply, per ft.	$0.31	$0.22½

Steam—Discounts from List

First grade....40–10% Second grade...50–5% Third grade....50–10–5%

RUBBER BELTING—The following discounts from list apply to transmission rubber and duck belting:

Competition..... 60–10–5% Best grade..... 60–5%

LEATHER BELTING—List price per ply, 12-in. wide, per lin. ft. $2.88

Grade	Discount from list
Medium	40–5%
Heavy	35%

RAWHIDE LACING { For cut, best grade, 50–10%, 2nd grade, 60%. For laces in sides, best, 41c. per sq.ft.; 2nd, 39c. Semi-tanned: cut, 50%; sides, 43c. per sq.ft.

PACKING—Prices per pound:

Rubber and duck for low-pressure steam, ½ in.	$0.90
Asbestos for high-pressure steam, ½ in.	1.80
Duck and rubber for piston packing	.90
Flax, regular	1.10
Flax, waterproofed	1.70
Compressed asbestos sheet	.90
Wire insertion asbestos sheet	1.30
Rubber sheet	.45
Rubber sheet, wire insertion	.70
Rubber sheet, duck insertion	.50
Rubber sheet, cloth insertion	.30
Asbestos packing, twisted or braided and graphited, for valve stems and stuffing boxes	1.40
Asbestos wick, ½- and 1-lb. balls	.70

PIPE AND BOILER COVERING—Discounts, New York warehouses, are as follows:

85% magnesia high pressure		60% off
For low-pressure heating and return lines	4-ply	70% off
	3-ply	72% off
	2-ply	74% off

PORTLAND CEMENT—New York, $2.40@$2.50 per bbl. without bags, in cargo lots delivered on job. Bag charge of 40c. per bbl.

STRUCTURAL STEEL—New York delivered price, 3 to 15-in. beams and channels and 3 to 6-in. angles, tees, and plates, all $2.08 per 100 lb.

COTTON WASTE—The following prices are in cents per pound:

	New York Current	Cleveland	Chicago
White	7.50@10.00	12.00	11.25
Mixed	5.50@ 9.00	9.00	8.00

WIPING CLOTHS—Jobbers' prices as follows:

	13½ x 13½	13½ x 20½
Cleveland	per M. $50.00	per M. $55.00
Chicago	per lb. 0.10	per lb. 0.13

LINSEED OIL—These prices are per gallon:

	New York	Cleveland	Chicago
Raw in barrels (5 bbl. lots)	$0.90	$1.07	$0.96

WHITE AND RED LEAD—Base price per pound:

	Red				White			
	Current		1 Year Ago		Current		1 Yr. Ago	
	Dry	In Oil	Dry	In Oil	Dry or In Oil	In Oil	Dry or In Oil	In Oil
100-lb. keg	12.50	14.00	13.00	14.50	12.50		13.00	
25- and 50-lb. keg	12.75	14.25	13.25	14.75	12.75		13.25	
12½-lb. keg	13.00	14.50	13.50	15.00	13.00		13.50	
5-lb. cans	15.50	17.00	16.00	17.50	15.50		16.00	
1-lb. cans	17.50	19.00	18.00	19.50	17.50		18.00	

RIVETS—The following quotations are allowed for fair-sized orders from warehouse:

	New York	Cleveland	Chicago
Steel ⅞ and smaller	60–5%	70%	60–10%
Tinned	60–5%	70%	4½c. per lb. net

Structural rivets, ½, ⅝, 1 in. diameter by 2 in. to 5 in. sell as follows per 100 lb.

New York.....$3.60 Chicago.....$3.10 Pittsburgh.....$2.40

Boiler rivets, same sizes:

New York.....$3.70 Chicago.....$3.20 Pittsburgh.....$2.50

REFRACTORIES—Prices in carlots:

Chrome brick, eastern shipping points	net ton	$42@45
Chrome cement, 40@45% Cr_2O_3	net ton	27@30
Chrome cement, 40@45% Cr_2O_3, in sacks	net ton	29.50@31.50
Magnesite brick: 9-in. shapes	net ton	56@58
Magnesite brick: 9-in. arches, wedges and keys	net ton	61.60@63.80
Magnesite brick: Soaps and spits	per ton	78.40@81.20
Silica brick: Chicago district	per 1,000	38@41
Silica brick: Birmingham, Ala.	per 1,000	42.00
Silica brick: Mt. Union, Pa.	per 1,000	33@35
Chrome ore crude, 45@50%	net ton	20@22
Magnesite dead burn (Imported)	net ton	30@33
Clay brick, 1st quality, 9 in. shapes, Pennsylvania	per 1000	33@36
Clay brick, 1st quality, 9 in. shapes, Ohio	per 1000	33@36
Clay brick, 1st quality, 9 in. shapes, Kentucky	per 1000	33@36
Clay brick, 2nd quality, 9 in. shapes, Pennsylvania	per 1,000	29@33
Clay brick, 2nd quality, 9 in. shapes, Ohio	per 1,000	29@33
Clay brick, 2nd quality, 9 in. shapes, Kentucky	per 1,000	29@33

BABBITT METAL—Warehouse prices in cents per pound:

	New York	Cleveland	Chicago
Fair grade	24.00	41.50	36.00
Commercial	11.00	16.00	9.00

COLD FINISHED STEEL—Warehouse prices are as follows:

	New York	Chicago	Cleveland
Round shafting and screw stock, per 100 lb. base	$3.35	$3.40	$3.30
Flats, square and hexagons, per 100 lb. base	3.85	3.90	3.80

BOILER SPECIALTIES—F. o. b. New York or Jersey City, discounts on list:

	Current
Copper ferrules	70%
Boiler flanges	60-10%
Boiler stay bolts	60%
Boiler patch bolts	10%
Boiler fitting-up bolts	60%
Pressed steel boiler lugs	10%
Pressed steel boiler hangers	10%

WROUGHT PIPE—The following discounts are to jobbers for carload lots on the latest Pittsburgh basing card:

BUTT WELD

Inches	Steel Black	Steel Galv.	Inches	Iron Black	Iron Galv.
1 to 3	71	58½	¼ to 1¼	44½	29½

LAP WELD

2	64	51½	2	39½	25½
2½ to 6	68	55½	2½ to 4	42½	29½
7 to 8	65	51½	4½ to 6	42½	29½
9 and 12	64	50½	7 to 12	40½	27½

BUTT WELD, EXTRA STRONG, PLAIN ENDS

1 to 1½	69	57½	¼ to 1¼	44½	30½
2 to 3	70	58½			

LAP WELD, EXTRA STRONG, PLAIN ENDS

2	62	50½	2	40½	27½
2½ to 4	66	54½	2½ to 4	43½	31½
4½ to 6	65	53½	4½ to 6	42½	30½
7 to 8	61	47½	7 to 8	35½	23½
9 to 12	55	41½	9 to 12	30½	18½

BOILER TUBES—Following are prices in New York warehouse of tubes manufactured according to specifications of the American Society of Mechanical Engineers:

Size	Lapweld Steel	C. C. Iron	Seamless Steel
1	.....	.....	$0.15
1¼	.....	.....	.1650
1½	.....	.....	.1411
1¾	$0.1654	$0.1881	.1530
2	.1305	.1607	.1332
2¼	.1491	.1836	.1521
2½	.1566	.2160	.1824
2¾	.....	.....	.....
3	.1916	.2668	.2097
3¼	.2066	.3078	.2253
3½	.2204	.3282	.2403
4	.2800	.4173	.3054

Tubes 2½ in. diameter, or smaller, over 18 ft. long, 10 per cent extra.

These prices are net per lineal foot based on stock lengths. If cut to special lengths, billing will be based on the entire stock lengths.

In addition to the above, standard cutting charges are as follows:

1¼ in. to 2 in. diameter, 5c. per cut. 2¼ in. diameter, 7c. per cut
2½ in. diameter, 6c. per cut. 3 in. diameter, 9c. per cut
3½ in. to 4 in. diameter, 10c.

ELECTRICAL SUPPLIES

ARMORED CABLE—Price per 1,000 ft.—5 per cent 10 days.

B. & S. Size	Two Cond. M Ft.	Three Cond. M Ft.	Two Cond. Lead M Ft.	Three Cond. Lead M Ft.
No. 14 solid	$ 41.00 (net)	$138.00	$164.00	$210.00
No. 12 solid	135.00	170.00	225.00	265.00
No. 10 solid	185.00	235.00	275.00	325.00
No. 8 stranded	285.00	375.00	520.00	500.00
No. 6 stranded	400.00	500.00	560.00	.....

From the above lists discounts are:

Less than coil lots	35%	10%
Coils to 1,000 ft	40%	20%
1,000 to 5,000 ft	45%	22%
5,000 ft. and over	45–5%	25%

CONDUIT, Price per 1,000 ft.; **ELBOWS AND COUPLINGS**, Per 100 pieces, f. o. b. New York, with 10-day discount of 5 per cent.

Size, In.	Conduit Black	Conduit Galvanized	Elbows Black	Elbows Galvanized	Couplings Black	Couplings Galvanized
¼	$46.58	$51.68	$ 9.65	$10.97	$4.75	$5.17
⅜	59.57	66.47	12.70	14.20	6.78	7.38
1	84.66	94.86	12.80	21.02	8.81	9.59
1¼	114.54	128.34	25.11	27.81	12.38	13.40
1½	136.95	153.45	33.48	37.08	15.29	16.55
2	184.26	206.46	61.38	67.98	20.38	22.06
2½	291.33	326.43	100.44	111.24	29.12	31.52
3	380.97	426.87	267.84	296.64	43.68	47.28
3½	485.76	540.96	591.48	655.08	58.24	63.04
4	597.32	662.72	683.55	757.05	72.80	78.80

CUT-OUTS—Following are net prices each in standard-package quantities:

CUT-OUTS, PLUG

S. P. M. L.	$0.08	D. P. D. B.	$0.26
D. P. M. L.	.13	T. P. S. B.	.25
T. P. M. L.	.19	T. P. D. B.	.38
D. P. S. B.	.15		

CUT-OUTS, N. E. C. FUSE

	0-30 Amp.	31-60 Amp.	60-100 Amp.
D. P. M. L.	$0.38	$1.20	$1.25
T. P. M. L.	.53	1.40	1 80
D. P. S. B.	.47	1.20	.....
T. P. S. B.	.88	2.00	.....
D. P. D. B.	.85	2.35	.....
T. P. D. B.	1.50	4.00	.....
T. P. to D. P. D. B.	1.00	2.75	.. ..

FLEXIBLE CORD—Price per 1,000 ft. in coils of 250 ft.:

No. 18 cotton reinforced heavy	$19.95
No. 16 cotton reinforced heavy	26.15
No. 18 cotton reinforced light	17.33
No. 16 cotton reinforced light	21.50
No. 18 cotton Canvasite cord	14.70
No. 16 cotton Canvasite cord	17.06

NATIONAL ELECTRIC CODE FUSES, NON-REFILLABLE—

250-Volt	Std. Pkg.	List	600-Volt	Std. Pkg.	List
3-amp. to 30-amp.	100	$.20	3-amp. to 30-amp.	100	$0.40
35-amp. to 60-amp.	100	0 30	35-amp. to 60-amp.	100	.60
65-amp. to 100-amp.	50	2.90	65-amp. to 100-amp.	50	1.50
110-amp. to 200-amp.	25	3.00	110-amp. to 200-amp.	25	2.50
225-amp. to 400-amp.	25	5.60	225-amp. to 400-amp.	25	5.50
425-amp. to 600-amp.	10	.50	450-amp. to 600-amp.	10	8.00

Discount: Less 1-5th standard package, 60%; 1-5th to standard package, 60–5%; standard package, 65–5%.

RENEWABLE FUSES, ENCLOSED—List price each:

Sizes	250-Volt List-Price	600-Volt List-Price	Std. Pkg. Quantity	Carton Quantity
1 to 30-amp.	$0.50	$1.10	100	10
35 to 60-amp.	1.00	1.25	100	10
65 to 100-amp.	2.00	3.00	50	5
110 to 200-amp.	4.00	5.00	25	5
225 to 400-amp.	7.50	11.00	25	1
450 to 600-amp.	11.00	16.00	10	1
450 to 600-amp.	11.00	16.00	10	1

REFILLS—

1 to 30	$0.30 ea.	$0.05	100	100
35 to 60	.05 ea.	.06	100	100
65 to 100	.10 ea.	.10	50	50
110 to 200	.15 ea.	.15	25	50
225 to 400	.30 ea.	.30	25	25
450 to 600	.60 ea.	.60	10	10

Discount Without Contract—Fuses:	
Broken carton	5%
Unbroken carton but less than std. pkg.	22%
Std. pkg.	40%
Discount Without Contract—Renewals:	
Less than std. pkg.	Net list
Standard package	40%
Discount With Contract—Fuses:	
Broken cartons	10%
Unbroken cartons but less than standard package	26%
Standard package	42%
Discount With Contract—Renewals:	
Less standard package	Net list
Standard package	42%

FUSE PLUGS, MICA CAP—

0-30 ampere, standard package	$2.40
0-30 ampere, less than standard package	2.75

LAMPS—Below are present quotations in less than standard package quantities:

Straight-Side Bulbs, Mazda B— Watts	Plain	Frosted	No. in Package	Pear-Shaped Bulbs, Mazda C— Watts	Clear	Frosted	No. in Package
10	$0.35	$0.40	100	75	$0.70	$0.75	50
15	.35	.40	100	100	.95	1.00	24
25	.35	.40	100	150	1.30	1.40	24
40	.35	.40	100	200	1.80	1.90	24
50	.35	.40	100	300	2.50	2.65	24
60	.40	.45	100				12
				500	3.75	3.95	12
				750	5.25	5.50	8
				1.000	6.00	6.35	8

Standard quantities are subject to discount of 10% from list. Annual contracts ranging from $150 to $300,000 net allow a discount of 17 to 40% from list.

PLUGS, ATTACHMENT—

	Each
Porcelain separable attachment plug	$0.17
Composition 2-piece attachment plug	.26
Swivel attachment plug	.12

RUBBER-COVERED COPPER WIRE—Per 1000 ft. f. o. b. New York.

No.	Solid Single Braid	Solid Double Braid	Stranded, Double Braid	Duplex
14	$ 6.56	$ 8.40	$10.02	$15.96
12	8.72	10.60	12.54	20 84
10	11.44	13.65	15.96	27.09
8	15.38	17.90	21.05	36.22
6	.....	.....	31.81	.....
4	.....	.....	42.94	.....
2	.....	.....	64.99	.....
1	.....	.....	86.83	.....
0	.....	.....	107.78	.....
00	.....	.....	127.89	.....
000	.....	.....	153.66	.....
0000	.....	.....	183.70	.....

SOCKETS, BRASS SHELL—

½ In. or Pendant Cap, Key Each	Keyless Each	Pull Each	¾ In. Cap, Key Each	Keyless Each	Pull Each
$0.33	$0.30	$0.60	$0.39	$0.36	$0.66

Less 1-5th standard package	15%
1-5th to standard package	30%
Standard package	45%

WIRING SUPPLIES—

Friction tape, ¾ in., less 100 lb. 34c. lb., 100 lb. lots	33c. lb
Rubber tape, ¾ in., less 100 lb. 34c. lb., 100 lb. lots	33c. lb
Wire solder, less 100 lb. 27c. lb., 100 lb. lots	21c. lb
Soldering paste, 2 oz. cans	$1.00 doz

SWITCHES, KNIFE—

TYPE "C" NOT FUSIBLE

Size, Amp.	Single Pole, Each	Double Pole, Each	Three Pole, Each	Four Pole, Each
30	$0.42	$0.68	$1.02	$1.36
60	.74	1.22	1.84	2.44
100	1.50	2.50	3.76	5.00
200	2 70	4.50	6.76	9.00
TYPE "C" FUSIBLE, TOP OR BOTTOM				
30	70	1.06	1.60	2.12
60	1.18	1.80	2.70	3.60
100	2.38	3.66	5.50	7.30
200	4.40	6.76	10.14	13.50

Discounts:	
Less than $25.00 list value	25%
$25 to $50 list value	30%
$50 list value or over	35%

New Plant Construction

PROPOSED WORK

Cal., Alameda—The city, W. E. Varcoe, Clk., will soon receive bids for furnishing and installing a deep well pump and machinery for the well in Franklin Park.

Cal., Presidio (San Francisco P. O.)—J. A. Wetmore, Acting Superv. Archt., Treas. Dept. Wash., D. C., received bids for pump house, equipment, etc., at the Marine Hospital here, from Engineering Structures Co., Call Bldg., San Francisco, $21,580; Schultz Constr. Co., 46 Kearny St., San Francisco, $23,500; Burnham Plumbing Co., 1220 Webster St., San Francisco, $23,900.

Cal., Sacramento—The California State Life Ins. Co., Forum Bldg., is having plans prepared for 10 story 100 x 100 ft. office building on 10th and J Sts. Estimated cost $1,000,000. Geo. C. Sellon & Co., Mitau Bldg., Stockton, Archts.

Cal., Sacramento—The Consumers' Ice & Cold Storage Co., 801 D St., plans additions and repairs to ice plant, work to be done by day labor. Estimated cost $19,500. Private plans.

Conn., Derby—I. J. Hoffman, Capitol Theatre, Ansonia, is receiving bids for 2 story 65 x 200 ft. stores and theater building, including steam heating and ventilating system, on Elizabeth St., here. Estimated cost $250,000. E. C. Horn & Son, 1476 Broadway, New York, Archts.

D. C., Wash.—The Dept. of Agriculture opened bids for furnishing a 300 hp. Sterling water tube boiler from Standard Eng. Co., 2129 I St., (a) $13,670 (b) $8,700 (c) $5,600 Casey & Co., 3207 14th St., N. W., (a) $13,900 (b) $8,304 (c) $5,596; McClintock Bros., (a) $14,020 (b) $11,659 (c) $2,361.

D. C., Wash.—Purchasing Agent, Post Office Dept., opened bids for a conveyor system from S. Olsen, Chicago, Ill., $55,000; deduct $950 for U. S. Belting; Mathews Gravity Carrying Co., Elwood City, Pa., $42,000 alternate $40,700 for Goodyear belting; Standard Conveyor Co., St. Paul, Minn., $53,812.80 plus $2,086 for automatic take-off, plus $1,353 for Crocker-Wheeler motors, deduct $349 if Westinghouse motors are used, deduct $1,428.30 if Goodyear belting is used.

D. C., Wash.—The War Dept., c/o Col. N. E. Wood, 2329 Munitions Bldg., 19th and B Sts., opened bids for a 3 story Army Medical School, including heating and ventilating systems, at the Walter Reed Hospital, here, (1) work complete from Boyle-Robertson Co., Evans Bldg., (1a) $431,557 (300 days) (1b) $429,557 (1c) $427,557 (1d) $427,557 (2) $2,809 (3) $4,000 (4) add $968 (5) $2,300 (6) $6,894 (7) $2,800 (8) $1,228 (9) $1,495 (10a) $12.15 (10b) $13.50 (10c) $1.50 (10d) $77 (10e) $2.16 (10f) $0.75 (10g) $0.40 (10h) $0.90; P. F. Gormerly, Union Trust Bldg., (1a) $429,853 except refrigeration, (2) $4,500 (5) $2,300 (6) $5,000 (7) $3,985 (8) $2,795 (9) $1,300; G. A. Fuller & Co., Munsey Bldg., (1a) $440,000 (240 days) (1b) $440,000 (1c) $440,000 (1d) $440,000 (2) $2,500 (3) $4,000 (4) add $1,000 (5) $2,400 (6) $5,000 (7) $3,985, $2,795 (9) $1,300. Noted June 13.

Fla., Lakeland—A Stock Corp., c/o B. M. Pulliam, Pres., is having plans prepared for an 8 story, 122 x 122 ft. hotel. Estimated cost $415,000. F. J. Kennard, 409 American Bank Bldg., Tampa, Archt.

Ga., Atlanta—The Capital City Building & Market Co., 603 Candler Bldg., plans to build 1 story 75 x 230 ft. market, including heating and 50 ton refrigerating plants, on Peachtree and Cain Sts. Estimated cost $250,000. G. C. Gibson, 603 Candler Bldg., Engr. Robert & Co., Archts.

Ill., Chicago—Grossman & Prosskauer, Archts., 127 North Dearborn St., is preparing plans for 7 story 100 x 150 ft. apartment building, including steam heating system, on 418 Diversey Parkway. Estimated cost $900,000. Owner's name withheld.

Ill., Chicago—A. V. Teisen, Archt., 4804 North Kedzie Ave., is preparing plans for 3 story, 125 x 180 ft apartment building, including steam heating system, on North Kedzie Ave. Estimated cost $250,000. Owners name withheld.

Ill., Chicago—Whitney & Williams, Archts., 122 South Michigan St., are revising plans for 3 story cafe, stores and dance hall building, including steam heating system, on Howard St. and Marshall Ave. Estimated cost $3,000,000. Owner's name withheld.

Ind., Fort Wayne—T. Keenan, c/o Anthony Hotel, 128 West Berry St., will receive bids until about July 15 for 15 story, 150 x 200 ft. hotel, including steam heating system, on Harrison and Washington Sts. Estimated cost $2,000,000. C. C. French, Plymouth Bldg., Cleveland, Ohio, Archt. W. B. Vetterhogg, Ft. Wayne, Engr. Noted Feb. 21.

Me., Waterville—The Sisters Hospital, College Ave., is receiving bids for 4 story addition to hospital, including steam heating and ventilating systems. Estimated cost $400,000. H. L. Rourke, 525 Hildreth Bldg., Lowell, Mass., Archt.

Md., Frostburg—The Frostburg Sewer Pipe Co., plans to build sewer pipe manufacturing plant on the north side of the city. Estimated cost $200,000. Engineer not announced.

Mich., Detroit—The Catholic Diocese of Detroit, c/o Rev. M. J. Gallagher, 1219 Washington Blvd., is having preliminary plans prepared for 2 story cathedral, including steam heating plant, on La Salle Blvd., between Webb and Lawrence Aves. Estimated cost $5,000,000. McGinnis & Walsh, 100 Boylston St., Boston, Mass.

Mich., Detroit—G. Klein, c/o A. Kahn, Archt., 1000 Marquette Bldg., plans to build an 8 story, 107 x 111 ft. hotel, including steam heating system on Pingree and 2nd Aves.

Mich., Detroit—The Studebaker Corp., Brush and Piquette Aves., will open bids about July 20 for a 4 story, 95 x 259 ft., factory addition, including steam heating system on Clark and Jefferson Aves. A. Kahn, 1000 Marquette Bldg., Archt.

Mich., Grand Rapids—The city, G. T. Wagner, Dir. Pub. Service will receive bids until July 13 for furnishing and erecting two 500 hp. boilers at the city pumping plant. R. E. Harrison, 351 Summit & Cherry Market Bldg., Toledo, Ohio, Engr.

Mich., Saginaw—The Secretary of the Bd. of Educ. will receive bids until July 12 for 2 story school building and first unit of detached power plant on the east side. Estimated cost $250,000. J. F. Beckbissinger, 114 South Jefferson St., Archt.

Miss., Natchez—The Natchez Ice Co. is in the market for cold storage and refrigeration machinery for proposed plant here.

Mo., Jefferson Barracks—J. A. Wetmore, Superv. Archt., Treas. Dept., Wash., D. C., received bids for the construction of the U. S. Veterans' Hospital, here, from J. Stewart & Co., Inc., 30 Church St., New York, $779,350; Westlake Constr. Co., Ry. Exch. Bldg., St. Louis, $787,000; W. M. Sutherland, 4412 Lindell Blvd., St. Louis, $814,900; mechanical equipment for the above, from W. Wilby, Selma, Ala., $307,976; S. Faith & Co., 2427 Pennsylvania Ave., Phila., Pa., $311,000; J. A. McBride, 1607 Olive St., St. Louis, $316,000.

Mo., Joplin—The Joplin Planing Mill is in the market for 15 or 20 hp. oil engine and belting.

N. Y., Ithaca—The Cornell University, M. Farrand, Pres., plans to build a recreation building and social centre. Estimated cost $1,000,000. Architect not announced.

N. Y., New York—Street & Smith, 79 7th Ave., will soon receive bids for 9 story publishing building at 147 West 15th St. H. O. Chapman, 334 5th Ave., Archt. and Engr.

N. Y., Rochester—The Ilex Mfg. Co., c/o M. Rosenblum, Prop., plans to build a 3 story, 85 x 220 ft. camera shutter plant on Portland Ave. Estimated cost $200,000. Architect not announced.

N. Y., Rochester—F. Jones, 123 Wilton Ave., is in the market for stationary gas engine, 8, 10 hp. or larger.

N. Y., Rochester—The Rochester Gas & Electric Corp., plans to change the heating system at steam driven plant and will heat large boilers with electricity, will install new type boilers to utilize surplus hydrauic power and convert same into steam. F. J. Howes, Chief Engr., c/o owner.

N. Y., Watkins—J. Rappleye is in the market for 16 hp. traction engine, used, in good condition.

N. C., Cumnock—The Rocky River Power & Light Co., W. E. Woody, Pres., High Point, will soon receive bids for the installation of a 500 kw. generator on Rocky River, also construction of 12,000 volt transmission line from Rocky River here. Mees & Mees, Charlotte, Engrs.

N. C., Wilmington—The Hotel Corp., R. Moore, Pres., is receiving competitive plans for hotel. Estimated cost $800,000. Noted May 2.

Ohio, Cleveland—The Bd. Educ., F. G. Hogen, Director, East 6th St. and Rockwell Ave., is in the market for boiler for the Orchard school.

Ohio, Cleveland—The Euclid 77 Co., c/o T. C. Minor, Archt., 9501 Wade Park Ave., is having plans prepared for two 4 story, 44 x 200 ft. apartments, including steam heating systems, 7711 Euclid Ave. Estimated cost $400,000.

Ohio, Cleveland—A. W. Harris, Archt., Discount Bldg., is receiving bids for 2 and 6 story apartment hotel, store and office buildings on Shaker Blvd., for the Riverview Imp. Co. Estimated cost $750,000.

Ohio, Cleveland—M. B. Scharfeld, Society For Savings Bldg., has had plans prepared for 12 story, 60 x 75 ft. commercial and office building, including steam heating system, on East 6th St. and St. Clair Ave. Estimated cost $500,000. Knox & Elliot, Engineers Bldg., Archts. and Engrs.

Ohio, Columbus—The Lloyd-Hoffman Co., E. Lloyd, Purch. Agt., 1148 Mt. Vernon Ave., is in the market for air compressor, and electric motor 15 or 20 hp.

Okla., Tulsa—The Oklahoma Public Service Co., Mayo Bldg. plans to build addition to power plant and install new equipment. Estimated cost $1,000,000. Sargent & Lundy, West Adams St., Chicago, Ill., Engrs.

Pa., Harrisburg—The Harrisburg Hotel Co., Third St., is having plans prepared for 13 story addition to hotel including steam heating system. W. L. Stoddart, 9 East 40th St., New York, Archt.

Pa., Millersburg—The Blue Mountain Electric Co. has purchased flour mill and plans to remodel into a power house, is in the market for power plant equipment to be used in generating electricity from water power.

Pa., Moon Run—The School District of Robinson Twp., J. W. Burkett, Secy., will receive bids until July 14 for furnishing and installing 60 cycle, a.c. 220 volt, single phase, electric pump with motor, about 2,050 ft. from the township high school.

Pa., Phila.—The Reformed Church of America and Marshall & Fox, Archts., 721 North Michigan Ave., Chicago, Ill., are receiving bids for 13 story, 120 x 150 ft. office building on 15th and Race Sts. Estimated cost $800,000.

Pa., Pittsburgh—The Donahoe Co., Inc., 31 Terminal Bldg., is having plans prepared for 3 story store and office building on Diamond St. Estimated cost $350,000. W. E. Snaman, Empire Bldg., Archt.

Tenn., Chattanooga—The Dixie Spinning Co., J. L. Lupton, Pres., plans to build a 10,000 spindle cotton and yarn mill. Estimated cost $500,000. Architect not announced.

Tenn., Memphis—The Mississippi River Comn., 1st and 2d districts, Customhouse, will receive bids until Juy 15 for furnishing and delivering Marine Type refrigerating and ice making plant.

Utah, Salt Lake City—The Continental Natl. Bank, 21 East 2nd St., and Natl. Bank of the Republic, Main St., are having plans prepared for 20 story bank and office building on Main and 2nd St. South. G. W. Kelham, Sharon Bldg., San Francisco, Cal., Archt.

Va., Scottsville—The Big Bend Power Co. plans large power development on the James River here, including large water reservoir, estimated cost $1,250,000. Quotations wanted on hydro electric machinery, pumping equipment, etc., to cost $250,000. Engineer not announced.

Wash., Seattle—The Dexter Horton Estate, New York Block, is having plans prepared for office building on 3d and Cherry Sts., first unit 12 story, 120 x 127 ft., second unit 14 story, 108 x 120 ft. Estimated cost $2,500,000. J. Graham, L. C. Smith Bldg., Archt.

W. Va., Charleston—J. P. Stotter, 4319 Prospect Ave., is having revised plans prepared for 3 story 100 x 253 ft. store and apartment building. Estimated cost $300,000. M. H. White, Schofield Bldg., Archt.

Wis., Hawkins—F. Prochaski, Milan, plans 60 x 100 ft. electric light plant, to furnish light, heat and power. Estimated cost $60,000. Architect not selected.

Wis., Madison—The Kennedy Dairy Co., 618 University Ave., plans to build 50 x 75 ft. ice manufacturing plant on Washington St. Estimated cost $40,000. Architect not selected.

Wis., Milwaukee—O. Brochman, 425 East Water St. and T. Saxe, 156 5th St., are having revised plans prepared for 5 story, 150 x 200 ft. business and office building on 6th and Grand Aves. Estimated cost $400,000. M. Tullgren & Sons, 425 East Water St., Archts.

Wis., Phillips—The Kneeland-McClurg Lumber Co., P. S. McClurg, Pres., plans to build 60 x 210 ft. saw mill. Estimated cost $200,000. Architect not selected.

Wis., Rice Lake—The city, O. Jenson, Clk., is having plans prepared for power plant, electric distribution system and other improvements. Estimated cost $75,000. J. H. A. Brahtz, 409 Metropolitan Opera House Bldg., St. Paul, Minn., Engr.

Wis., Sheboygan — The Lake Shore Constr. Co., Engrs., will soon receive bids for boiler and low pressure steam heating plant and equipment for store and office building, here, for the I. Friede Clothing Co., Wausau.

Wis., Wauwatosa—The Milwaukee Bd. Trustees, W. L. Coffey, Mgr., will receive bids until July 13 for refrigeration machinery.

Wyo., Basin—The city and Midwest Refining Co., Greybull, plan the construction of 66 mi. transmission line from Cody Dam to Greybull to here. Estimated cost about $50,000.

B. C., Ocean Falls—The Pacific Mills, Ltd., Vancouver, plans the construction of hydro-electric generating plant in connection with saw and pulp paper mill here. Estimated cost $450,000 to $500,000. Engineer not announced.

N. S., Halifax—The Premier Paper & Power Co., P. H. Moore, Mgr. Dir., 91 Hollis St., plans to build a large paper mill 35 miles from here to include 2 hydraulic turbines to generate about 3,200 hp. and 4 water wheels to run grinders. Estimated cost $1,500,000. G. F. Hardy, 309 Broadway, New York, Engr.

Ont., Belleville—The Belleville General Hospital plans to build boiler house and laundry, including horizontal tubular boilers. Estimated cost $25,000. Architect and engineer not appointed.

Ont., Hamilton—The Kanadda Biscuit Co., 185 West Ave., N., plans to build 2 story, 80 x 400 ft. factory, including steam heating system. Estimated cost $400,000. B. H. Prack, 60 Bay St., Toronto, Archt.

Ont., Welland — The Welland Cotton Mills, Ltd., L. B. Spencer, Secy., 60 Merritt St., W., is having plans prepared for cotton mills on Queen and Duncan Sts. Estimated cost about $200,000. W. J. Westaway Co., 38 Main St., West Hamilton, Archts. Noted June 20.

CONTRACTS AWARDED

Cal., Los Angeles—The Bd. Educ., Security Bldg., has awarded the contract for 2 story high school on Boyle Heights to K. R. Bradley, 1925 East 16th St., $307,157; steam heating system to F. J. Donnelly, 1031 North Wilton Pl., $21,960.

Cal., San Francisco—The city and county of San Francisco have awarded the contract for the construction of the Pulgas Tunnel, Contract 85, Hetch Hetchy project, to the Grant-Smith Co., Balboa Bldg., $683,050.

Cal., San Luis Obispo—The city council has awarded the contract for pump, 500 gal. per minute, against head 125 lb. capacity, to the San Luis Hardware & Implement Co., $2,105. Noted June 20.

Col., Santa Barbara—The Mission City Hotel Co., A. E. Bern, Pres., has awarded the contract for 4 story, 115 x 180 ft. hotel on State St., to the United Construction Co., Baker-Detwiler Bldg., Los Angeles. Estimated cost $250,000.

Conn., Thompsonville—The Bigelow Hartford Carpet Co., Tariff St., has awarded the contract for 1 story, 82 x 142 ft., dyehouse, 113 x 145 ft. worsted mill and 111 x 227 ft. Jacquard mill addition, to L. E. Locke & Sons Co., South Union St., Lawrence, Mass. Estimated cost $1,000,000. Steam heating and ventilating systems to be installed.

Ill., Chicago—H. F. Bloomfield, 118 North LaSalle St., has awarded the contract for 3 story apartment building on 70th Place to Welch-McCarthy Constr. Co., 56 West Washington St. Estimated cost $650,000. Steam heating system will be installed.

Ill., Chicago—E. Edelman & Co., c/o D. S. Klafter, Archt., 64 West Randolph St., has awarded the contract for 2 story, 150 x 250 ft. factory for the manufacture of automobile accessories, on Logan Blvd. and Holly Ave. to Paschen Bros., 111 West Washington St. Estimated cost $250,000. Steam heating system will be installed.

Ill., Chicago—F. P. Jackson, c/o L. C. Bouchard, Archt., 30 North Dearborn St., has awarded the contract for 10 story, 102 x 230 ft. apartment building on 3318 Sheridan Rd. to Schmidt Bros. Constr. Co., 22 East Huron St. Estimated cost $1,500,000. Steam heating system will be installed.

Ill., Chicago—The Shelbourne Apt. Hotel Corp., c/o W. P. Doerr, Archt., 28 East Jackson Blvd., has awarded the contract for 10 story, 140 x 170 ft. apartment hotel on Hyde Park Blvd. and 53d St., to the Mueller Constr. Co., 179 West Washington St. Estimated cost $2,000,000. Steam heating system will be installed.

Ia., Storm Lake—The city has awarded the general contract for pumping station to W. F. Park & Co., $6,523, plumbing and heating, to L. S. Slugosch, $685. Noted Jan. 24.

Me., Portland—The city, c/o Deering High School Comn., has awarded the contract for 3 story, 70 x 400 ft. high school, at Presumpscot Park, to F. T. Ley & Co., Inc., 499 Main St., Springfield, Mass.; steam heating and ventilating systems to Willey & Calhoun. Estimated cost $611,857. Noted May 9.

Mich., Ann Arbor—The Bd. of Regents, University of Michigan, has awarded the contract for 4 story, 189 x 248 ft. engineering shops on East University St. to H. G. Christman & Co., 315 Stevens Bldg., Detroit, about $700,000. One 40 hp. and two small motors will be installed. Noted Apr. 11.

Minn., St. Paul—The city has awarded the contract for furnishing and erecting two 250 hp. water tube boilers at McCarron's pumping station, for the Water Dept., to J. G. Robertson, 2542 University Ave., $25,813. Noted June 13.

Minn., St. Paul—The Washington Foundry Co., H. H. Orme, Pres., Eagle and Washington Sts., has awarded the contract for 1 story, 71 x 166 ft. and 2 story, 22 x 100 ft. foundry buildings to F. J. Romer Constr. Co., 190 Ramsey St. Is in the market for air compressor, electric motors and other equipment.

Mont., Red Lodge—The city has awarded the contract for a clear well, 2 pressure regulators, 13,020 lin.ft. of 6, 8 and 10 in. and 8,900 lin.ft. of 8 or 10 in. c.i. pipe to McLaughlin & O'Neil, Livingston, $63,106.

Neb., Lincoln—Sanitary Dist. 1, Lancaster Co., has awarded the contract for pumping equipment for the sewage disposal plant to Kent Engineering Co., 504 First Natl. Bank Bldg., Omaha, $10,000. Noted July 4.

N. J., Manasquan—The Bureau of Yards & Docks, Navy Dept., Wash., D. C., has awarded the contract for a radio compass station, here, to N. B. Smyth, Inc., 417 East 34th St., New York, $4,900 (60 days). Spec. 4663.

N. J., New Lisbon—The state has awarded the contract for well, pump, pumphouse and tower for the State Colony for Feeble Minded, to N. A. K. Bugbee Co., 206 Hanover St., Trenton, $13,500.

N. Y., Long Island City—The Bd. Educ., 500 Park Ave., New York, has awarded the contract for Public School No. 19 on 41st and Lake Sts., also Public School No. 69 on Polk Ave., to the Putnam Constr. Co., 303 5th Ave., New York, $418,000 and $414,000 respectively.

N. Y., New York—The Bd. Educ., 500 Park Ave., has awarded the contract for Public School No. 80 on East 120th St. to the T. A. Clarke Co., 122 Livingston St., Brooklyn, $487,300. Noted Sept. 6, 1921.

N. Y., Niagara Falls—The Bd. of Educ. has awarded the contract for furnishing and installing new heating plant for high school and junior high school to the Matthews Plumbing & Heating Co., $73,225.

Okla., Wewoka—The Bd. of Trustees has awarded the contract for waterworks improvements consisting of filtration plant, intake and repairing dam to Yates & Gallamore, Cushing, $38,524. Noted June 27.

Pa., Altoona—The Confederated Home Abattoir Corp., 123 Willow Ave., has awarded the contract for 3 story, 80 x 140 ft. meat packing plant on Eldorado Rd., to L. R. Mathiew Constr. Co., 302 Grant Ave. Noted May 23.

Tex., Dallas—The Hella Temple, 400 South Harwood St., has awarded the contract for hospital to the Munn Constr. Co., S. W. Life Bldg. Estimated cost $250,000.

W. Va., Peach Creek—The Chesapeake & Ohio Ry., 9th and Main Sts., Richmond, Va., has awarded the contract for power house, round house and machine shop here to J. E. Nelson & Son, 3240 South Michigan Ave., Chicago, Ill., $250,000.

W. Va., Ronceverte—The West Penn Power Co., A. M. Lynn, Pres., West Penn Bldg., Pittsburgh, Pa., has awarded the contract for addition to power plant to J. G. White Engineering Corp., 43 Exchange Pl., New York. Estimated cost about $500,000.

Wis., Ganotown—The Bd. of Educ., c/o A. H. Zander, Clk., has awarded the contract for 3 story, 240 x 419 ft. high and grade school to T. Stark & Co., 130 Muskego Ave., Milwaukee, $379,000; heating and ventilating systems to W. S. Patterson & Co., Appleton, $58,636.

Wis., Milwaukee—The Sewage Comn., J. H. Fowler, Secy., 508 Market St., has awarded the contract for superstructure of main boiler house and boiler plant on Jones Island to Paul Riesen's Sons, 1018 Humbold Ave., $211,890. Noted June 20.

Ont., Toronto—The Farmers Dairy Co., Walmer Rd., has awarded the contract for 2 story, 106 x 120 ft. addition to dairy building to the Jackson-Lewis Co., Ltd., Ryrie Bldg., $150,000. Steam heating system and dairy equipment for pasteurizing, etc., to be installed.

H. T. Hilo—The Bureau of Yards and Docks, Navy Dept., Wash., D. C., has awarded the contract for radio towers, here, to W. Will, Hilo, $30,986. Spec. 4666. Noted June 13.

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